

Proceedings of:

The Thirteenth Annual Meeting

The American Society for Precision Engineering

October 25-30, 1998

Hyatt Regency St. Louis at Union Station

St. Louis, Missouri

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

*ASPE — The American Society for Precision Engineering
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Preface

This book comprises the proceedings of the 1998 Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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Welcome Note

The American Society for Precision Engineering celebrates its Thirteenth Annual Meeting this year in St. Louis, Missouri. The remarkable growth and success of the Society is reflective of the increasing importance of precision engineering in a wide variety of fields of endeavor, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into a premier international forum for the exchange of ideas and presentation of research results relating to precision engineering problems involving mechanical parts, electronic components, materials and material processing, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

This year's meeting continues the tradition of offering two full days of tutorials presented by some of the foremost precision engineers and scientists in the world. Many participants cite the tutorial program as one of the most valuable features of the meeting. The technical sessions and poster presentations offer the latest research results in the areas of ultraprecision machining, metrology, controls, precision grinding, micro-positioning, material processing, design, precision transducers, surface profilometry, machine tool metrology, and error compensation. The commercial exhibit will provide attendees with the opportunity to view and discuss the latest precision engineering equipment, products, and services that are commercially available. The open forum will give attendees the opportunity to express their opinions and pose questions for discussion by the group, enhancing the exchange of information.

The conference organizing committee is proud to present the program for the Thirteenth Annual Meeting of ASPE. We welcome your participation.

Welcome to St. Louis.

1998 ASPE Meeting Organizing Committee

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Hy Dinh Tran, University of New Mexico

David L. Trumper, Massachusetts Institute of Technology

Program

1998 ASPE Annual Meeting

<i>Time</i>	<i>Sun., Oct. 25</i>	<i>Mon., Oct. 26</i>	<i>Tues., Oct. 27</i>	<i>Wed., Oct. 28</i>	<i>Thurs., Oct. 29</i>	<i>Fri., Oct. 30</i>
8:00			Continental Breakfast			
9:00	Tutorials	Tutorials + Executive Forum	Technical Session 1	Technical Session 4	Technical Session 7	Technical Tours
10:00			Break	Break	Break	
11:00			Technical Session 2	Technical Session 5	Technical Session 8	
12:00			Lunch + Committee Meetings	Awards Lunch + Business Meeting	Lunch + Roundtable Discussions	
1:00						
2:00	Tutorials	Tutorials + Executive Forum	Technical Session 3	Technical Session 6	Technical Session 9	
3:00			Break	Break	Break	
4:00			Poster Session 1 & Exhibits Hospitality Hour	Poster Session 2 & Exhibits Hospitality Hour	Technical Session 10	
5:00						
6:00			Commercial Session	Topical Open Forums		
7:00		Keynote Address				
8:00		Welcome Reception	Dinner	Open Evening		
9:00						

1998 Annual Meeting

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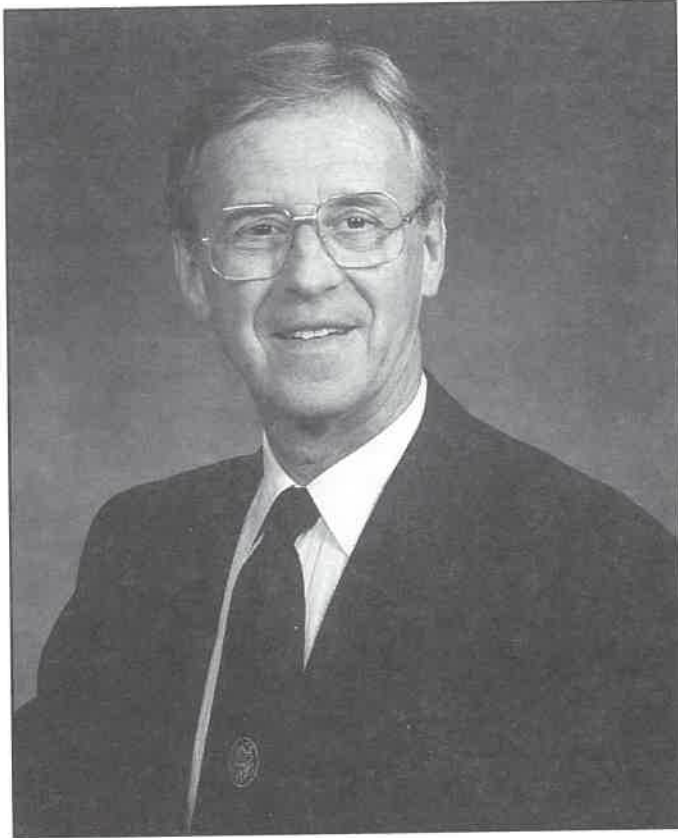
1998 Scholarship Recipients

*Edward Acworth, Stanford University
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1998 Lifetime Achievement Award

Patrick A. McKeown



The 1998 ASPE Lifetime Achievement Award is presented to Professor Patrick A. McKeown in recognition of his contributions to the development and promotion of the arts and sciences of precision engineering.

Following military service in the British Royal Engineers, Pat McKeown began his technical career as a student apprentice at the Bristol Aircraft Company, England. He studied aircraft design, production engineering and industrial administration at Cranfield before joining the Societe Genevoise d'Instrumente de Physique (SIP) in 1956. At SIP he worked on a variety of high precision measuring machines, including the photo-electric line standard comparator installed at B.I.P.M. in Sevres, France. In 1965, he was appointed Technical Director of SIP's UK subsidiary, taking a responsibility for manufacture of jig borers and the design and build of other high precision specialty machines.

In 1968, Pat returned to Cranfield as Chief Engineer (later Director) of the Cranfield Unit for Precision Engineering (CUPE), which became self-supporting (based on industrial contracts) within three years. He was appointed to a personal chair at Cranfield, and in 1976, launched an academic department that paralleled the ac-

tivities of the industrial unit. Under Pat's leadership, CUPE expanded rapidly, developing both machine systems – such as the electronic gearbox and the CUPROC controllers – as well as a range of special purpose high precision machine tools and measuring machines. Two subsidiary companies – Cranfield Moulded Structures (epoxy granite) and Cranfield Precision Systems (controls) – emerged before CUPE itself was transformed into Cranfield Precision Engineering, Ltd. in 1987.

Throughout his career, Pat has been a tireless promoter of precision engineering principles and practice, teaching and lecturing throughout the world. He played a major role in the launch of "Precision Engineering," now ASPE's journal, was instrumental in launching the International Precision Engineering Seminar Series in 1981, and is a Charter Member of ASPE. He is an active member of C.I.R.P. (The International Institution for Production Engineering Research), having served as its President in 1988, and is a Fellow of the UK Royal Academy of Engineering and a Charter Fellow of the Society of Manufacturing Engineers (SME). His many honors include SME's Frederick W. Taylor Research Medal, the Order of the British Empire, and honorary doctorates from the Universities of Connecticut and Cranfield.

Since his (notional) retirement in 1995, Pat McKeown has continued to teach and to promote the arts and sciences of precision engineering. Most recently, his long campaign to launch a European Society for Precision Engineering and Nanotechnology has come to fruition, with its first conference scheduled for May 1999.

Keynote Speaker

Duncan T. Moore

Monday, October 26, 6:30 p.m.

Session Chair: Chris J. Evans, National Institute of Standards and Technology

Dr. Moore was confirmed by the U.S. Senate in the fall of 1997 for the position of Associate Director for Technology in The White House Office of Science and Technology Policy. In this position he works with Dr. Neal Lane, President Clinton's Science Advisor, to advise the President on U.S. technology policy, including the Next Generation Internet, Clean Car Initiative, new construction materials, and NASA. Dr. Moore is on leave from his position as the Rudolf and Hilda Kingslake Professor of Optical Engineering at the University of Rochester. Previously, from 1995 until the end of 1997, he served as Dean of Engineering and Applied Sciences at the University. He also served as president of the Optical Society of America, a professional organization with 12,000 members worldwide, in 1996.

The PhD degree in optics was awarded to Dr. Moore in 1974 from the University of Rochester. He had previously earned a master's degree in Optics at Rochester and a bachelor's degree in Physics from the University of Maine.

Dr. Moore has extensive experience in the academic, research, business, and governmental arenas of science and technology. He is an expert in gradient-index optics, computer-aided design, and the manufacture of optical systems. He has advised nearly 50 graduate thesis students. In 1993, Dr. Moore began a one-year appointment as Science Advisor to Senator John D. Rockefeller IV of West Virginia. He also chaired the successful Hubble Independent Optical Review Panel organized in 1990 to determine the correct prescription of the Hubble Space Telescope. In addition, Dr. Moore is the founder and former president of Gradient Lens Corporation of Rochester, NY, a company which manufactures the high-quality, low-cost Hawkeye boroscope.

Of note, in addition, is that the Honorable Duncan Moore was elected to the National Academy of Engineers in February 1998. He has been the recipient of the Science and Technology Award of the Greater Rochester Metro Chamber of Commerce (1992), the Distinguished Inventor of the Year Award of the Rochester Intellectual Property Law Association (1993), and the Gradient-Index Award of the Japanese Applied Physics Society (1993). He also received an Honorary Doctor of Science degree from the University of Maine in 1995.

"Making Science and Technology Policy — An Engineer's Viewpoint"

Dr. Moore will give his impressions after ten months of working at the White House's Office of Science and Technology Policy; how policy is made, and how the public can have an impact. The current budget situation for the fiscal years 1999 and 2000 will be discussed and how it will affect the science, engineering and technology communities.

Executive Forum

Monday, October 26, 8:00 a.m.

Session Chair: John H. Bruning, Tropel Corporation

For the second year we will be holding the Executive Forum. The purpose of this event is to bring together business executives in our industry and provide them with useful and contemporary information to help better plan and manage their business. Selected topics include novel manufacturing processes using lasers and magnetic fluids; new precision grinding methods, optical and machined surface metrology, to name a few. There will also be ample opportunity during the Forum for attendees to interact with industry experts and peers in a casual environment.

Commercial Session

Tuesday, October 27, 5:00 p.m.

Session Chair: Irving F. Stowers, Lawrence Livermore National Laboratory

The commercial session will take place on Tuesday afternoon, in a special session where company representatives are invited to make brief presentations on new products associated with precision engineering. Conference participants will have the opportunity to hear about limitations of performance of existing approaches, new approaches relevant to precision engineering, and advances that can be expected in the future.

Open Forum

Wednesday, October 28, 5:00 p.m.

Session Chair: John A. Patten, University of North Carolina at Charlotte

The Open Forum was created to encourage dissemination of new and significant results, as well as observations on precision engineering subjects. This year we will hold three concurrent Open Forums on specific topics.

One Forum will be chaired by John A. Patten of the University of North Carolina at Charlotte and will deal specifically with a "Summary and follow-up on the Spring Topical meeting on Silicon Machining." A Summary session was held at the April 1998 meeting to discuss what is known and what is not known about machining silicon. An appreciation of impediments to further progress in machining silicon and understanding the phenomena involved in these machining processes resulted from this summary session, and will be subject of this open forum.

Further topics will be developed as well. All interested conference participants can sign up at the Conference Registration Table for short, informal presentations.

Technical and Poster Paper Index

The technical program for the 1998 ASPE Annual Meeting contains over 170 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster session. Authors will be present to discuss their work on Tuesday, October 27, from 3:30 to 5:00 p.m., and again on Wednesday, October 28, from 3:30 to 5:00 p.m.

Session I Metrology I

Tuesday, October 27, 1998, 8:30 a.m. - 10:00 a.m.

Session Chair: Dan Luttrell (Corning Inc.)

Welcome Address

D. Luttrell (Corning Inc.)

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R. D. Day, R. Dickerson, C. Maggiore, P. Brooks, R. Springer, B. Carter, D. Hatch (Los Alamos National Laboratory), P. E. Russell (North Carolina State University), and T. Woodward, T. Stark (Materials Analytical Services) 29
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*Extended abstract not available at press time.

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*Extended abstract not available at press time.

**Abstract unavailable at press time.

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Technical Papers



A S P E

Development of Ultra-Precision 3D-CMM Based on 3-D Metrology Frame

Hisashi SHIOZAWA, Yasushi FUKUTOMI, Tamotu USHIODA and Susumu YOSHIMURA
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1-6-3 NISHI-OHI, SHINAGAWA, TOKYO, 140, JAPAN

1. INTRODUCTION

Optical element surface figures are widely measured through interferometric techniques. Generally, these techniques require the use of master optics, which have simple geometrical surfaces. However, the testing of off-axis or aspherical optical elements traditionally requires the use of null optics such as a computer-generated hologram (CGH). On the other hand, coordinate measuring machines (CMM's) can be used to easily obtain optical element profiles without the use of null optics. Until now, the limited uncertainty of CMM's has forced them to be used only in a complementary role to interferometric techniques. We report on the development of an ultra-precise CMM, "Super-MASTER", with uncertainty comparable to that of interferometric techniques.

2. DESIGN CONCEPT

The target repeatability of Super-MASTER is less than $5\text{nm}(1\sigma)$ in measuring volume of $400(X) \times 400(Y) \times 100\text{mm}(Z)$, using the contact probe. Super-MASTER's Metrology Loop, that is, the closed loop between the stylus contact point and the work-piece surface, is shown in Figure 1. Errors occurring in this loop, such as the result of physical deformation, lead directly to measurement errors. Errors in the metrology loop can be the result of internal error: imperfections in the reference mirrors, uncertainty in the wavelength of the measurement laser, etc., as well as external disturbances: ground vibration, temperature variation, etc. Accounting for these factors in the error budget, computer analysis and preliminary experiments have yielded the following design targets:

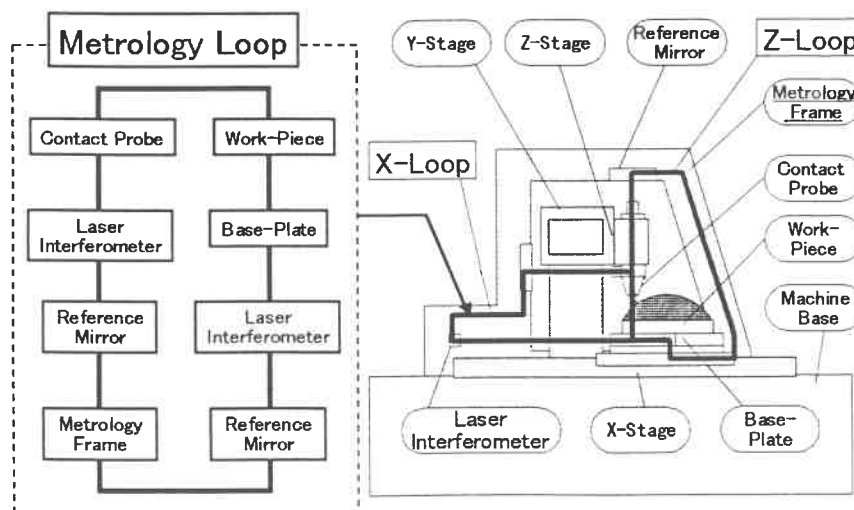


Fig.1: Metrology Loop

- a. Metrology Frame: $< 1\text{nm}$ of static and dynamic deformation.
- b. Measurement Laser: $< 1\text{nm}(1\sigma)$ of error in the overall measuring area.
- c. Contact Probe: $< 0.5\text{mN}$ of contact force.
- d. Vibration: $< 2 \times 10^{-5}\text{m/s}^2$ of acceleration on the machine base.
- e. Atmospheric Thermal Stability: $< \pm 0.02\text{C}^\circ$ around the machine.

3. CONSTRUCTION

Super-MASTER, shown in Figure 2, is constructed with horizontal X and Y-axis, and a vertical Z-axis, with a measuring volume of $400(\text{X}) \times 400(\text{Y}) \times 100\text{mm}(\text{Z})$. The work-piece is mounted on the X stage, and a contact probe system is attached on the lower end of the Z stage, which is, in turn, mounted on the Y stage. These stages are supported by air bearings and driven by friction drive units (FDU's), reducing mechanical vibration.

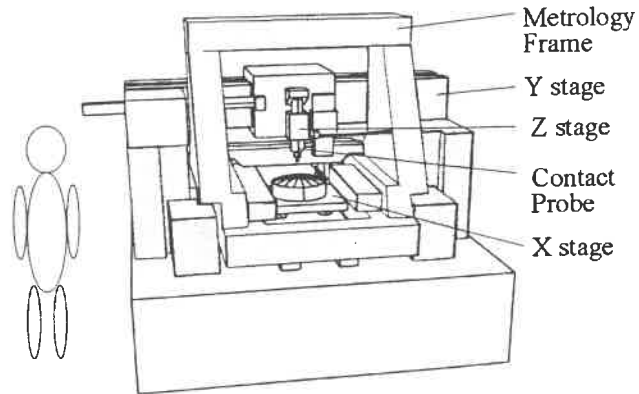


Fig.2: Ultra-precise CMM, "Super-MASTER"

4. 3-D METROLOGY FRAME

The metrology frame of Super-MASTER is independently mounted to the machine base by 3 kinematic, elastic support systems, immersed in damping oil. These systems isolate the frame from ground vibration and reduce the deformation of the machine base due to stage motion. The damping oil viscosity and the stiffness of the elastic supports have been optimized by structural analysis. In order to increase the resonant frequency of the metrology frame structure, these support systems are located in a plane that includes the frame center of gravity.

Six, laser interferometer reference mirrors are mounted on the metrology frame. The location and orientation of the mirrors and the laser passes are shown in Figure 3. The probe base-plate position is monitored in 5 degrees of freedom (excluding Z-axis rotation), while the work-piece base-plate monitors all 6 degrees of freedom. The result is that the relative position between the work-piece and the stylus is completely determined.

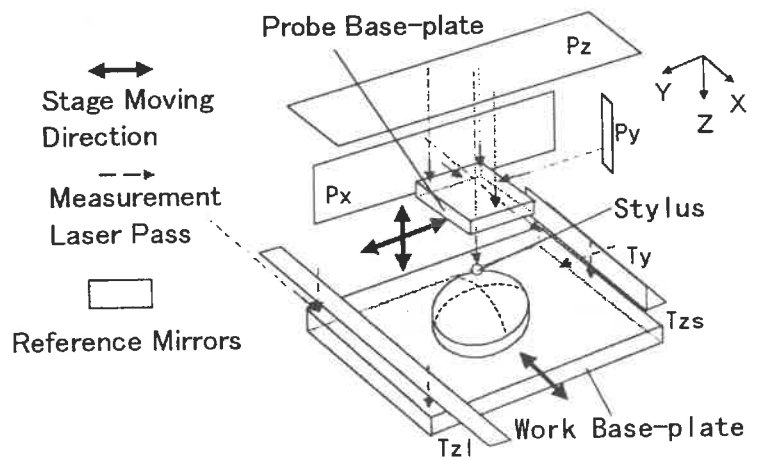


Fig.3: Mirrors and Laser Pass

5. VACUUMED LASER PASSES

Long laser passes in Super-MASTER are enclosed in vacuum bellows with internal pressure kept at 0.1Pa, eliminating the influence of air turbulence. The force on the bellows incurred due to external air pressure is isolated from the metrology frame.

The experimental apparatus to confirm the effect of the laser vacuum pass and the result of experiment are shown in Figure 4. The air gap in this system corresponds to the gap between mirrors on the metrology frame and vacuum bellows windows of Super-MASTER.

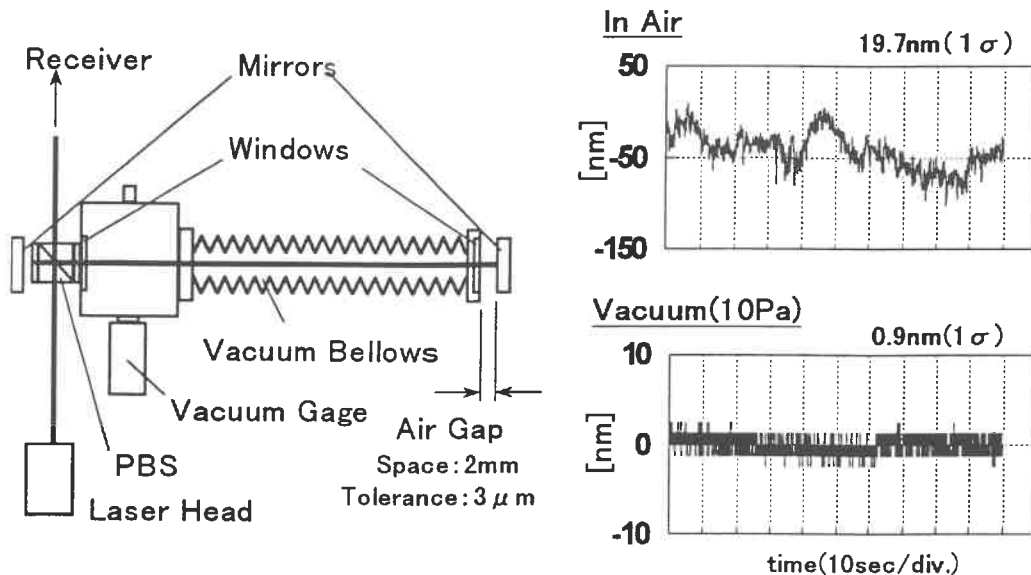


Fig.4: Experimental Apparatus for Vacuum Laser Pass

6. LOW FORCE CONTACT PROBE

The probe rod is supported by air bearing, while the rod weight is canceled by a small coil spring (Fig.5). The Z position of the probe rod, and the XYZ position and tilt angle of the probe base-plate are measured by laser interferometers.

By the control of the relative displacements between the probe rod and the probe base-plate, the contact force between the stylus and the work-piece is kept at 0.5mN, reducing deformation of the probe unit and wear on the stylus.

7. MEASUREMENT RESULTS

Measurements were made on a plane with a lateral dimension of 400mm, and a concave,

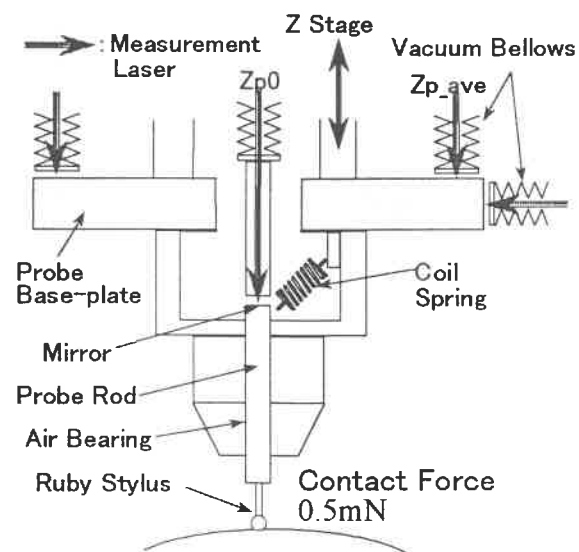


Fig.5: Low Force Contact Probe

spherical lens with a diameter of 400mm, a radius of curvature of 257mm, and a height of 80mm. The difference between two data sets, obtained under identical conditions, one day apart, is shown in Figure 6. This data shows the repeatability of Super-MASTER was 2.1nm (1σ).

Using the above-mentioned concave spherical lens, the bias error was estimated to be 9.5nm (1σ) through statistical methods. The distortion of the measuring coordinate due to the reference mirrors imperfection of flatness and orthogonal setting, and an uncertainty of the probe stylus radius were mainly included in this data. A total uncertainty of 19.5nm (2σ) was determined from these results.

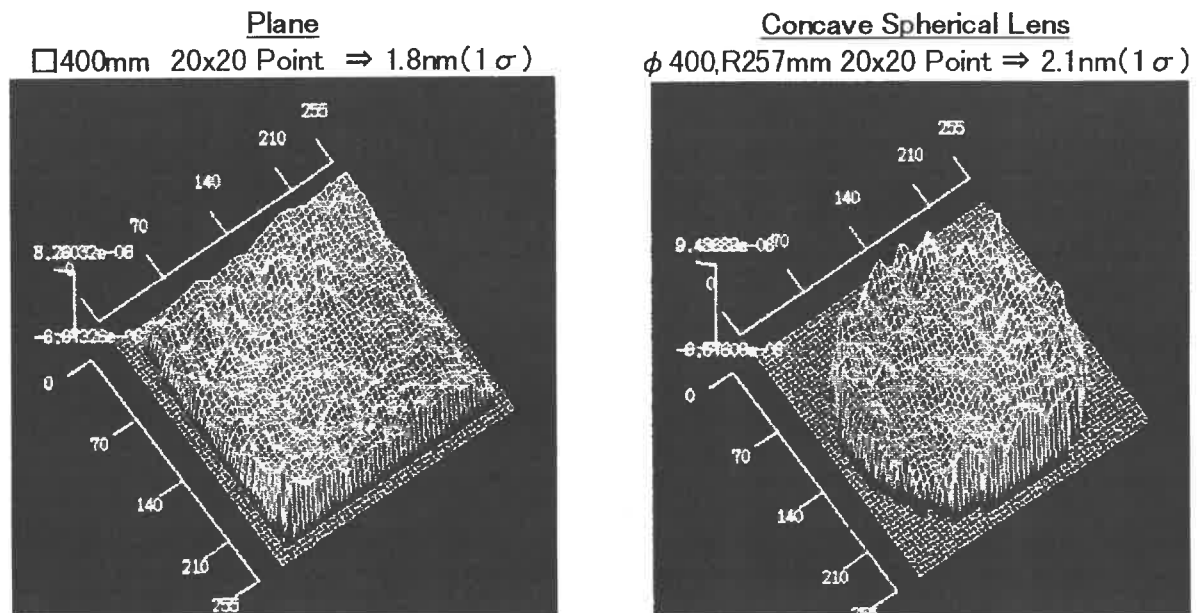


Fig.6: Difference between two data sets, under identical conditions, one day apart

8. CONCLUSIONS

As we have shown in this paper, Super-MASTER, an ultra-precise CMM, has proven capable of measuring surfaces with repeatability of 2.1nm(1σ), and uncertainty of 19.5nm (2σ), in measuring volume of 400(X) x 400(Y) x 100mm(Z). This performance compares with that of interferometric techniques. Simple geometrical figures, a plane and a spherical surface were used in this evaluation. The measuring uncertainty for aspherical surface figures should be similar because the uncertainty of CMMs, such as Super-MASTER, does not depend on the objective figures, but rather the objective dimensions. In the future, we plan to reduce the bias error through statistical methods.

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Two-dimensional stage self-calibration: Role of symmetry and invariant sets of points¹

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ABSTRACT. Modern semiconductor lithography stages are capable of reproducibility at nanometer scales, and nanometer-scale measurement errors can have serious impact on wafer yield in fabrication facilities. Yet at this time there is no two-dimensional measurement artifact or procedural standard of comparable and verified accuracy that can be used to calibrate these tools. The fundamental problem of high-precision positional metrology therefore is this: how can you make a standard measurement artifact that is as accurate as the precision of the tool to be calibrated if you do not already have an accurately calibrated machine for making the artifact? A rigorous method for using an ebeam stage to calibrate itself, known as *stage self-calibration*, was published in 1985 and has since been refined in various ways and has demonstrated the potential for accuracy throughout the workspace in the range of 10-20 nm.

Self-calibration methods can be understood best by considering certain point sets left invariant by the movements of the measurement artifact. The simplest illustration is afforded by a method for locating the midpoint of a straight wire that takes advantage of the fact that the midpoint of the wire is left unmoved by in-place reversal of the wire. More generally, when a rigid two-dimensional object (artifact) is moved for remeasurement from one position to another in the workspace of the measuring machine, some special set of points on the object, remains unchanged. In particular, certain Bravais lattices are invariant under their defining translations and rotations, and their shapes are completely determined once their symmetries are specified. It is invariance properties like these that make reversal and self-calibration methods possible and link the theory of self-calibration to crystallographic groups.

This paper briefly illustrates the ongoing evolution of stage self-calibration methods from a geometric perspective, using classical reversal methods as illustrations. Its point of departure is a recent paper by Evans, Hocken and Estler that classified 18 reversal and self-calibration techniques and discussed some of the symmetries that characterize the classic reversals.² While that paper emphasized algebraic symmetries, this paper demonstrates that geometric invariants are the governing features that underlay reversals and self-calibration methods. That geometric perspective can be exploited to derive improved techniques and explicit error bounds for high-accuracy self-calibration. It will be of interest to process engineers and metrologists seeking insights and methods for improving the accuracy of high-precision stages.

Keywords. Self-calibration, reversals, separation of errors, positional metrology, geometric invariants, orbits, Bravais lattices.

¹ This talk is based on an invited paper read at *The 41st International Conference on Electron, Ion and Photon Beam Technology & Nanofabrication*, published in the *Journal of Vacuum Science and Technology B*, 15(6), Nov/Dec 1997.

² Evans, Chris J., Hocken, Robert J., and W. Tyler Estler, "Self-Calibration: reversal, redundancy, error separation, and 'absolute testing,'" *CIRP Annals*, Vol 45/2, 1996, pp617-634..

Rapid Precision Volumetric Metrology
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ABSTRACT

In this paper, we will discuss the design and implementation of a non-scanning volumetric metrology system. This system is defined as being able to make precise location measurements within a large volume of, for example, 1 m^3 . The system, referred to as OpticalGPS™ (U.S. Patent No. 5.793.483), is analogous to the well known satellite based, microwave, L-band Global Positioning System (GPS), but instead uses range coded optical beams generated from low power, eyesafe laser sources to establish the location and orientation of an object in three-dimensional space. The system inherently measures absolute range, is less subject to signal fade problems, and it utilizes mature technologies such as RF electronics and laser diode transmitters. Although this technology approaches the position resolution of interferometric techniques, it avoids the critical alignment issues associated with interferometry and is therefore easier and less expensive to implement in manufacturing applications. In addition, this technique surveys the entire measurement volume simultaneously and as a result doesn't require the use of an expensive azimuth-and-elevation gimbal scanner to track the object. This alone reduces the cost and complexity below existing volume metrology technologies.

The design realizes very high position resolution, at the $45 \text{ } \mu\text{m}$ (3σ) level at measurement rates greater than 60 Hz. The critical challenge has been achieving commensurate precision. Factors, which will be discussed, that influence precision and accuracy include computation errors, stray signals, electronic and optical as well as environmental effects. We will illustrate

how the system is calibrated to establish orientation of the object being positioned and a reference frame or coordinate system. The requirement for numbers of transmitters and receivers, fields-of-view, and redundancy to overcome problems of vignetting and loss-of-signal.

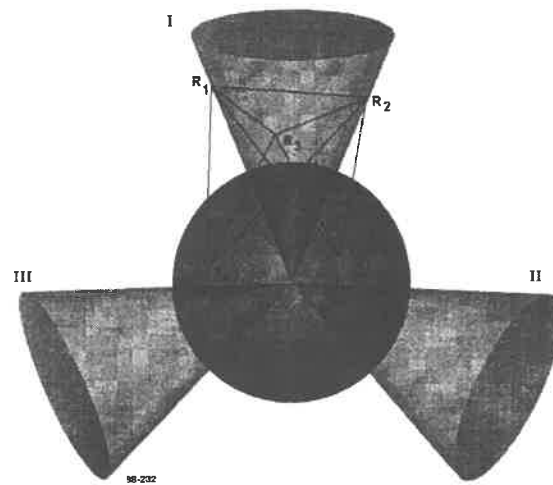
INTRODUCTION

Volumetric metrology, where the location of a point or points are measured on an object within some volume is a commonly occurring requirement for purposes of quality control, reverse engineering and calibration or control of the end-effector position of an industrial robot. A good example of such a capability is the coordinate measurement machine or CMM using a mechanical touchprobe tied to a x, y, z reference frame. Their relative slowness because of the sequential nature of the measurements have focussed attention on other ways to make these measurements including optical ones. A good example of an optical volumetric measurement capability was a system first demonstrated by (Lau, et al) at NBS, now NIST and apparently commercialized by a number of companies including the SMX Corporation, Kenneth Square, PA. This system measures the vector to a point on an object in 3D space by measuring the distance to typically a retroreflector mounted on the object using a distance interferometer and the elevation, azimuth angles via an angle tracking mirror. The measurements are however, still sequential and the distance measurement, being interferometric has a very short ambiguity range, $\lambda/2$ e.g., $0.3 \mu\text{m}$ so loss of signal due to beam interruption and e.g., target or atmospheric speckle, turbulence needs to be strictly avoided. The high cost of laser interferometry is also factor limiting their use. A concept that probably not surprisingly comes to mind to make 3-D volumetric measurements is to borrow a page from GPS. GPS is a satellite based navigation system, in precise orbits, with each satellite sending coded range information, in the form of a low repetition rate pseudorandom, e.g., P-code with a large ambiguity range, each satellite having a common epoch time, clock reference. Using for example the reconstituted carrier phase, displacement, e.g., on the ground $< 1 \text{ mm}$ can be resolved. Microwave signals, L-band $\approx 1.5 \text{ GHz}$ do not penetrate walls of building so GPS cannot be used for indoor positioning or navigation. However, one could design an indoor microwave GPS using small, so called pseudolite transmitters mounted on the ceiling, walls, etc. Such a design has been demonstrated

a Stanford University (Cohen, Rock) to control a small free floating robot. Microwave signals because of their relative long wavelength e.g., 19 cm for GPS results in low antenna gains. They are also, because of the long wavelength subject to multi-path effects, reflections e.g., off metal surfaces and microwave hardware is relatively expensive. A design under development and demonstrated at Visidyne is a mixture of optical techniques and microwave hardware. It closely parallels GPS, but it uses intensity modulated optical beams, at frequencies similar to GPS, ≈ 1 GHz from multiple optical transmitters to fill a given volume with range coded light and then uses large field of view receivers covering the same volume. By decoding the range information, a point, points are located in 3-D space. The feasibility of such a design results from the availability of small, low cost laser diodes with high output powers > 100 mwatts that can be intensity modulated in the GHz range and receivers, detectors with high radiometric performance, sensitivity. Standard microwave electronics from e.g., the cellular phone field can then be used to amplify and process the modulation to arrive at ranges, distances.

System Description

The overall concept for an Optical GPS™ is illustrated in Figure 1, including a set of performance requirements. Figure 2 shows how the ranges are determined by measuring the delay, phase shift of intensity modulated light from small low power laser diodes producing a photon density wave in space with a wavelength $\lambda = c/f_m$. The modulation frequency f_m can either be the carrier frequency of 1 GHz or a lower rate, amplitude or phase modulation superposed on the carrier. As shown in Figure 1, the system is configured as an inverted GPS with multiple receivers mounted on a reference frame and a single transmitter for each point to be located. As in GPS accuracy can be further improved by adding sources at calibrated, surveyed locations within the measurement volume thereby constituting a Differential or D-Optical GPS™.



System Specifications - OGPS™
 Measurement Envelope - 1 Mdia Sphere
 Positional Resolution - < 10μm
 Positional Precision - 45 μm (3σ) R₁
 Vector Angle Precision - < 3 mrad (3σ)
 Measurement Rate - > 30 per sec

Receivers R₁, R₂, and R₃ in a Precise Stable
 Metrology Frame
 R₂ - Transmitters, T₁, T₂ located at points to be
 measured
 I, II, III, Alternate views, in case of obscurations

Figure 1. Design for an Optical GPS

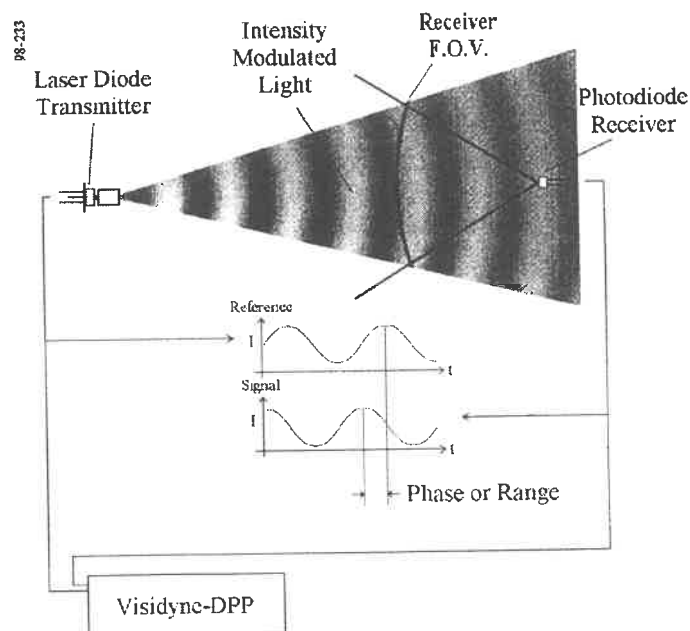


Figure 2. Using Intensity Modulated Light to Measure Range

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OPTO-TACTILE SENSOR FOR MEASURING SMALL STRUCTURES ON COORDINATE MEASURING MACHINES

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INTRODUCTION

The increasing miniaturization in the manufacture of mechanical structures frequently confronts the classical tactile coordinate measuring technique with unsolvable tasks. The smallest probing ball diameter which can be realized today for commercial probing systems is 0.2 mm. This probing ball size leads, however, to considerable handling problems. Also, due to the great elastic effects in the probe shaft, the accuracy of the measurements decreases strongly.

For the measurement of small structures, in most cases, optical procedures are preferred. Optical coordinate measuring machines which are combined with an efficient image processing unit allow mainly two-dimensional structures to be satisfactorily investigated. But when real workpieces are measured by optical methods, frequently difficulties arise:

- Technical surfaces differ strongly in their optical properties (colour, reflectance, roughness)
- Transparent materials (e.g. acrylic glass for glass fibre connectors) cannot be measured
- Unequivocal edge detection is frequently impeded by chamfers and material faults
- Fully three-dimensional geometries are in most cases difficult or impossible to measure
- Blind holes in particular are hard to illuminate and to image

These problems become particularly clear when very small boreholes are measured ($< 0,5$ mm) such as in wire draw dies, screens or injection nozzles for diesel or spark ignition engines, which are of great economic importance. To enable structures in the submillimeter range to be measured by mechanical probing, numerous attempts are made to further miniaturize mechanical probing systems [1,2,3]. All these designs depend, however, directly or indirectly on the sensing of force effects. At PTB, a probing system has now been developed, which combines the advantages of optical and tactile measuring techniques. This system will in the following be described.

FUNCTIONAL PRINCIPLE

The functional principle is based on the determination of the probing ball position by an optical imaging system and a CCD camera (Fig. 1). The "probe pin" used is an optical fibre whose tip is provided with a spherical probing element which is illuminated through the fibre.

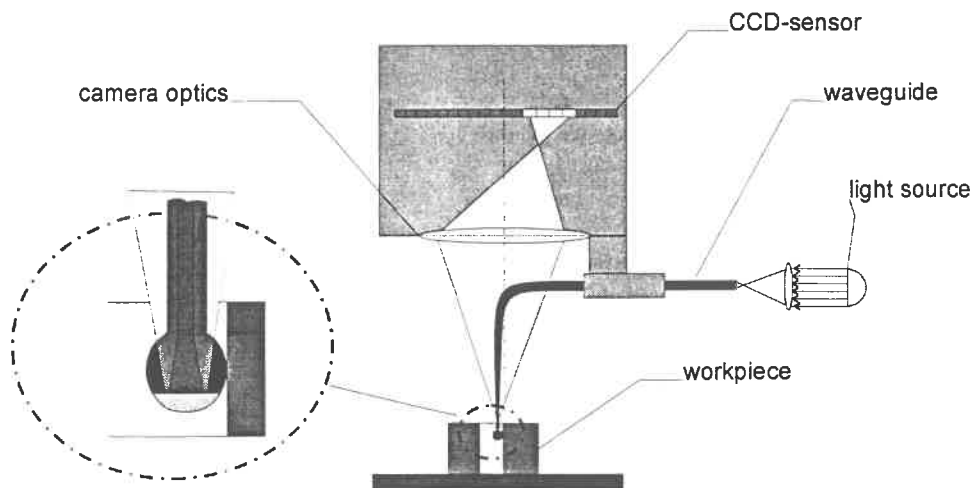


Fig. 1: Principle of the opto-tactile micro-probe

The optical fibre is mounted in the optical axis of the system such that the probing element is kept in the focussing plane of the optical system. The backscattered light of the ball is imaged on the CCD sensor as a bright light spot, whereas the rest of the fibre remains largely invisible to the camera as it is outside the focussing plane. When the probing element comes into contact with the workpiece surface (i.e. when shifting with respect to the camera), the light spot changes its position on the sensor. This change in position can be evaluated with subpixel accuracy.

Using an optical magnification of 10 and a CCD sensor with a pixel size of $10\text{ }\mu\text{m}$, a scale of $1\text{ }\mu\text{m}$ per pixel is obtained. Under optimum conditions, by subpixeling, a lateral resolution of the measuring system of better than $0,05\text{ }\mu\text{m}$ can be achieved when imaging the probing element over more than 50 pixel on the circumference. The resolution of the system which can actually be achieved is, however, also strongly dependent on the imaging quality of the light spot (Fig. 2).

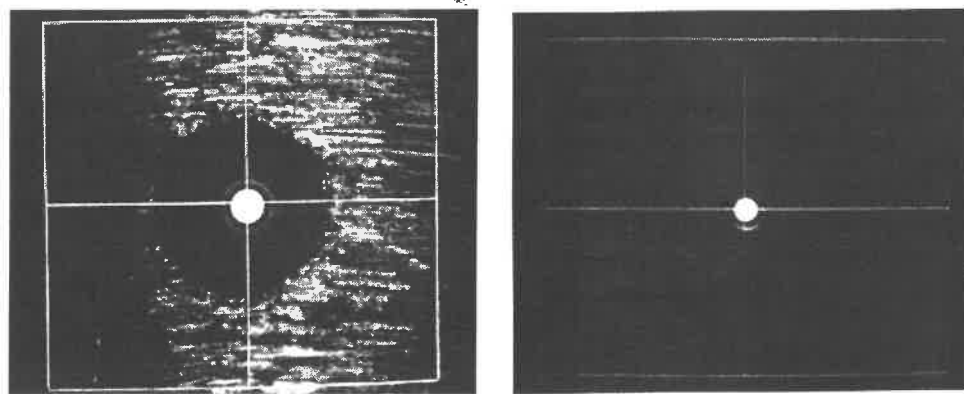


Fig. 2: Image of probing element and borehole in the CCD camera with additional incident light for visual positioning (left) and exclusive illumination of the probing element for measurement (right)

It is a fundamental advantage of this system that no force transmission between probing element and sensor is necessary. This eliminates all influences by elastic or plastic deformation of the probe shaft. The shaft thus can be extremely thin: Up to now, shaft diameters of $20\text{ }\mu\text{m}$ and probing ball diameters of $40\text{ }\mu\text{m}$ have been realized. All experiments so far have shown, that adhesive effects occurring between workpiece surface and probing element (frequently supported by a fine water film adhering to the surfaces) do not influence the measurement result.

DESIGN

For the functional optimization of the system, the following points in particular were of importance:

- low form errors of the probing element
- good optical properties of the probing element (strong reflection/backscattering)
- good mechanical properties of the glass fibre (resilience, breaking strength)
- ease of adjustment of the glass fibre in the image field and in the focus of the camera
- high optical quality of the imaging optics
- suitable calibration strategy
- adapted image processing software

Different procedures were tried for manufacturing the probing element: thermoplastic shaping of a spherical element and gluing of a glass sphere to a glass fibre end (Fig. 3).

Problems with the detection of the sphere position are caused by the light falling from the illuminated probing element on the surface of the workpiece where it is reflected or scattered. This light can result in softening the contrasts of the light spot or even in a real mirror image of the probing element forming on the workpiece surface. It is, however, possible to reduce these effects by providing a reflecting coating and/or separating probing element and target.

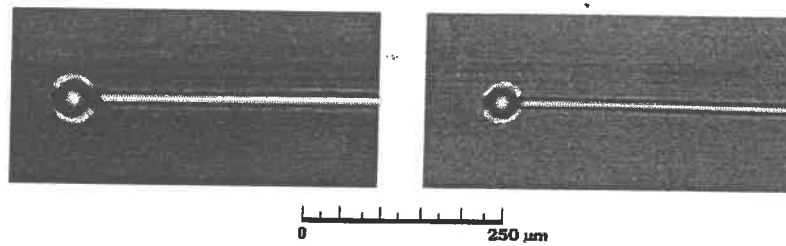


Fig. 3: Micrographs of probing elements: variant molten from the glass fibre (left) and glued (right)

The mechanical properties of the glass fibre can be optimized by skilfully proportioning diameters and lengths and by the selection of the procedure for the thermal deformation of the glass fibre. In manufacture, we have up to now started from the quartz glass multimode optical fibres with a core diameter of 50 μm and an outer diameter of 125 μm . These are heated and drawn to a diameter of less than 20 μm to reach the required order of magnitude.

The accuracy of adjustment does not immediately effect the accuracy of measurement: The CCD-sensor is the reference system, and the adjustment needs to ensure only that the probing element is brought into the visible range of the sensor.

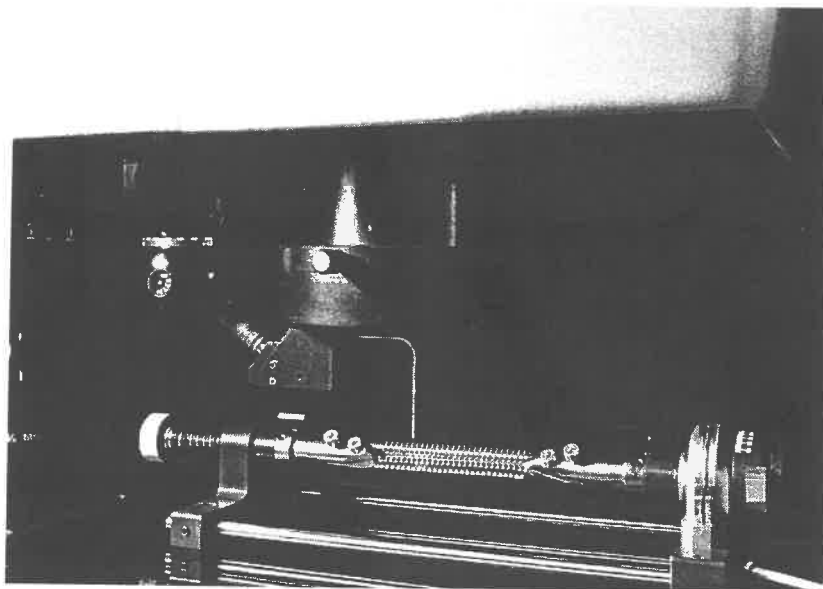


Fig. 4: Opto-tactile micro-probe mounted on an optical CCM [4]

An important point for the practical application of the probing system is a suitable calibration procedure. The radius of the probing element must be determined, and the reference to other sensors in the system must be defined. As for a classical tactile system, calibration can be made through a calibration sphere. This may have the usual diameter of 25 to 30 mm. The system as available up to now is, however, sensitive only in the direction of the image plane, probing of the sphere may take place only close to the sphere equator. Thus the reference in the third spatial direction is given with a higher uncertainty.

EXPERIMENTAL RESULTS

In the following, results will be described which were obtained at PTB with a probing ball diameter of 60 μm and a shaft diameter of approx. 35 μm . The free shaft length of this system is approx. 10 mm but the maximum penetration depth into boreholes only 1 to 5 mm according to the diameter of the boreholes: If the probe penetrates a borehole to a greater depth, the light passage from the probing element to the evaluation optics is not sufficient for ensuring reliable evaluation by the image

processing unit. To demonstrate the capabilities of the system, a complete cylinder measurement on a micro-borehole 0.8 mm deep and 214 μm in diameter was carried out. Fig. 5 shows the measured form errors of the borehole in a lateral plot.

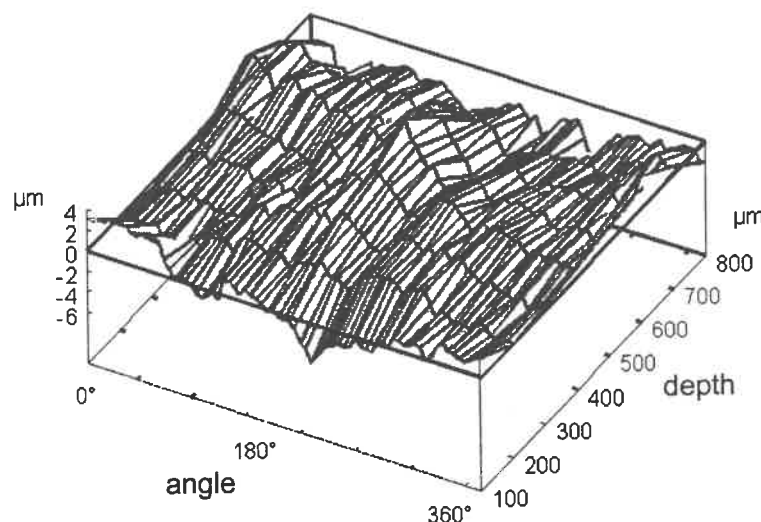


Fig. 5: Form errors of a borehole 214 μm in diameter and 850 μm deep

The standard deviation reached for unidirectional probing at PTB is appr. 0.1 μm ; It is to be expected that for the two-dimensional case comparable results can be achieved, since it seems to be limited only by the geometric errors of the machine, the optical components and the image processing.

Depending on the design, probing forces of only 1-10 μN are obtained, which also allows strongly resilient workpieces to be measured without scarcely any deformation.

AREAS OF APPLICATION

Potential areas of application of the system presented are all fields where conventional probing systems are utilized, especially for measuring tasks where force sensing systems can no longer be used because of their size. Besides the small boreholes described above, these are in particular

- external threads
- small gears
- clock components
- connectors (microelectronics and optical fibre technology)
- other micromechanical components
- very resilient workpieces (small wall thickness or elastic material)

The combined use with conventional optical measuring procedures appears to offer particular advantages. In Fig. 4 a now commercially available system is shown, where the probe can be automatically rotated into the field of sight of the camera, thus both optical and opto-tactile measurements can be performed in the same reference coordinate system.

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SUBSURFACE DAMAGE IN METALS INDUCED BY THE PLOUGHING COMPONENT OF PRECISION MACHINING PROCESSES

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Introduction

To achieve the dimensional tolerances required in ultraprecision machining, very small depths of cut are often used in the final machining steps. It has been shown [1] that for normal ultraprecision machining conditions the effective depth of cut that produces the surfaces is less than 25 nm. At this depth of cut, ploughing of the material becomes the dominant mechanism in producing the part's surface. In multiple point machining techniques, such as polishing and grinding, ploughing is a significant mechanism in producing the final part's surface.

Since ploughing plays such an important role in the production of the final surface for these widely used methods, it was decided to investigate the role that ploughing has on the surface integrity of metals. Microploughing experiments were conducted to acquire this information.

Description of Experiments and Results

A single crystal diamond pyramidal indenter was used to plough the (001) and (111) surfaces of large single crystals of gold and iridium. These two materials were chosen because they both have face centered cubic (FCC) lattices, they are both chemically inert (which means that no surface oxides will be present to obscure the data), and they represent the extremes in mechanical properties for FCC metals. The ploughing force normal to the surface was measured as the grooves were generated by an electronic microbalance. The ploughing depth was measured with an atomic force microscope (AFM) after the grooves were produced. The indenter geometry was measured using both scanning (SEM) and transmission electron microscopy (TEM).

Calculations revealed that the average yield stress of the gold increased with depth of cut from 10 MPa (1,300 PSI) to 200 MPa (25,800 PSI) for depths ranging from 100 nm to 1000 nm. At depths above 1000 nm the yield stress remained constant at 200 MPa. The average yield stress for iridium increased from 100 MPa (13,000 PSI) to 3000 MPa (387,000 PSI) as the depth of cut increased from 20 nm to 100 nm. Above 100 nm the yield stress remained constant at 3000 MPa.

After generating the grooves, some of them were cross sectioned for TEM observation. Since the cross sectioning area had to be located to an accuracy of 500 nm, the focused ion beam was used to perform the sample preparation.

Selected area diffraction measurements of the TEM samples in gold showed induced lattice rotations that ranged from 2° near the groove bottom to over 20° in the raised material on the side of the groove. Electron diffraction analyses showed regions near the groove bottom with high angle grain boundaries which had been rotated as much as 26° from the ploughing direction. These observations are consistent with a severely worked material as indicated by the yield stress calculations.

Acknowledgments

This work was funded by the Department of Energy under contract W7405-ENG-36; this funding is gratefully appreciated. At least one author, RDD, is grateful to LORD for affording him the opportunity to gain a better understanding of His fixed laws of nature through this work.

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ULTRAPRECISION MILLING OF COMPLEX SURFACE GEOMETRIES

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The fabrication of complex surface geometries is one of the main future trends in ultraprecision technologies. Tendencies towards real 3D-structures and sculptured surfaces are forced by the integration of multiple functions in one product. In addition, molds and dies for optical applications and products with microstructured surfaces require more and more a non-conventional multiaxis machining.

To date, ultraprecision machine tools are mainly build in a lathe-type design, limiting the producible workpiece shape to rotational symmetric geometries. To enlarge the product spectrum, a three axis ultraprecision milling machine was designed and built at the IPT in Aachen (*fig. 1*). This milling machine is equipped with two air bearing spindles and linear air bearing guides. The y-guide is provided with a high speed milling spindle, permitting ultraprecision milling operations with optical surface quality to be carried out on the machine. The air-bearing milling spindle is build as a hybrid design, combining the spiral groove technology with an externally pressurized bearing type. At a maximum rotational speed of 100.000 rpm, a radial stiffness of 30 N/ μ m is achieved in radial direction with a total error motion of below 100 nm. This spindle was developed in corporation with Philips, Zeiss and Cranfield Precision.

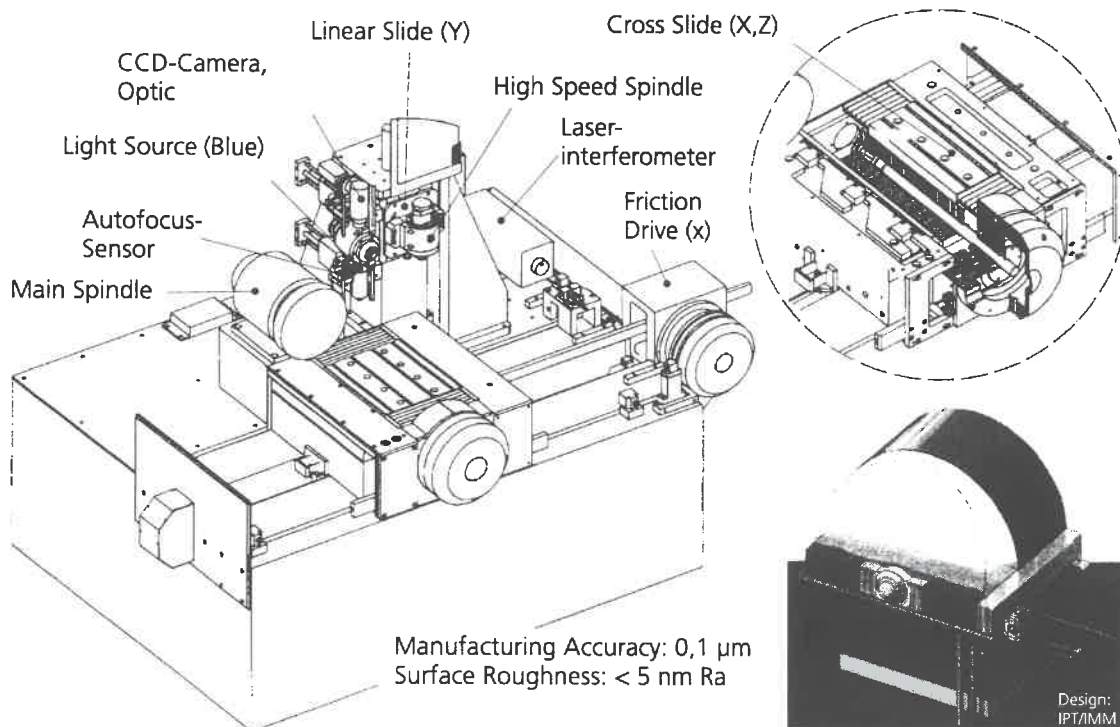


Fig. 1: Ultraprecision Milling Machine 'UPM' (Fraunhofer IPT)

To measure the current position, a laser interferometer system is integrated into the linear slides. In combination with high precision DC-torque-motors and servo systems, a position accuracy in the submicron-range has been achieved. The geometry of the rotating milling tool is directly measured using an optical system, mounted on the y-slide. Thus, a chucking error of the tool can be compensated by the control system.

In addition to this three-axis machine tool, a five-axis ultraprecision milling machine is currently build up in a joint project of IPT and the company Kugler. This production machine is equipped with hydrostatic linear guides to improve the stiffness and damping characteristics. Two of the slides are driven by linear motors to avoid a mechanical coupling between feed drive system and moving components, high resolution linear scales are integrated into these axis. Two additional rotary axis enable a real 5-axis milling to be carried out on the machine. The high speed spindle, built at the IPT, is identical to the above described hybrid spindle. The deployment of a precision spindle with such high rotational speed reduces the machining time , thereby increasing the economic efficiency of this technique. This 5-axis machine is build for milling operations as well as for precision grinding.

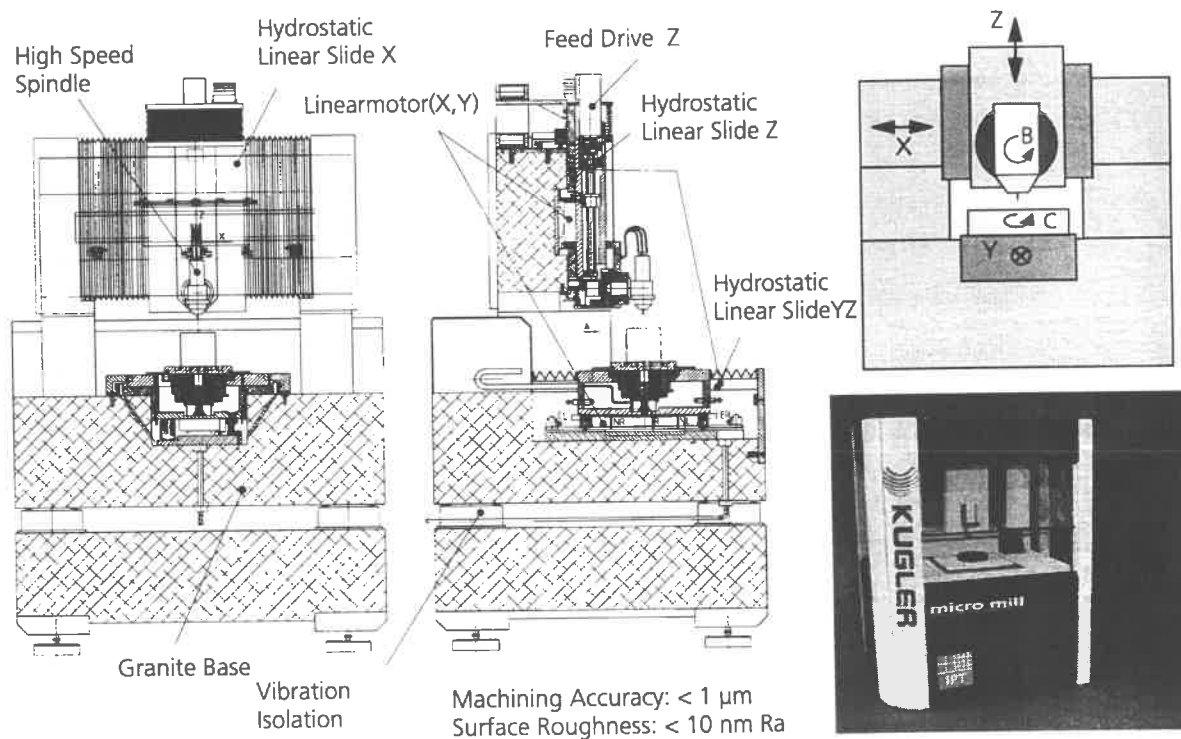


Fig. 2: Ultraprecision Milling Machine 'Micro Mill' (Fraunhofer IPT/Kugler)

Part geometry and surface quality is influenced on the one hand by the machine characteristics and on the other hand, to a decisive degree, by the milling tool. At present, only the diamond can

be sharpened to the required level of accuracy. Cutting edge sharpness and cutting edge roundness are crucial to the manufacturing quality of the workpiece.

In fig. 3, three types of natural diamond milling tools are shown. The endmill with a diameter of 300 μm (1) is used for the manufacturing of micro-grooves. An adjustable diamond ball-endmill with a radius of 0,5 mm is also exposed (2). The ball-endmill allows the fabrication of sculptured surface geometries with optical surface qualities. In addition, a fly-cutting milling tool is shown (3). This tool permits the fabrication of micro-grooves with minimum diameters of about 40 μm . These precision diamond tools require certain clamping devices to position the cutting edge in the micron and submicron range. High accuracy draw-in collet chucks and adjustable holders in combination with optical tool measurement systems have to be integrated into the milling machine.

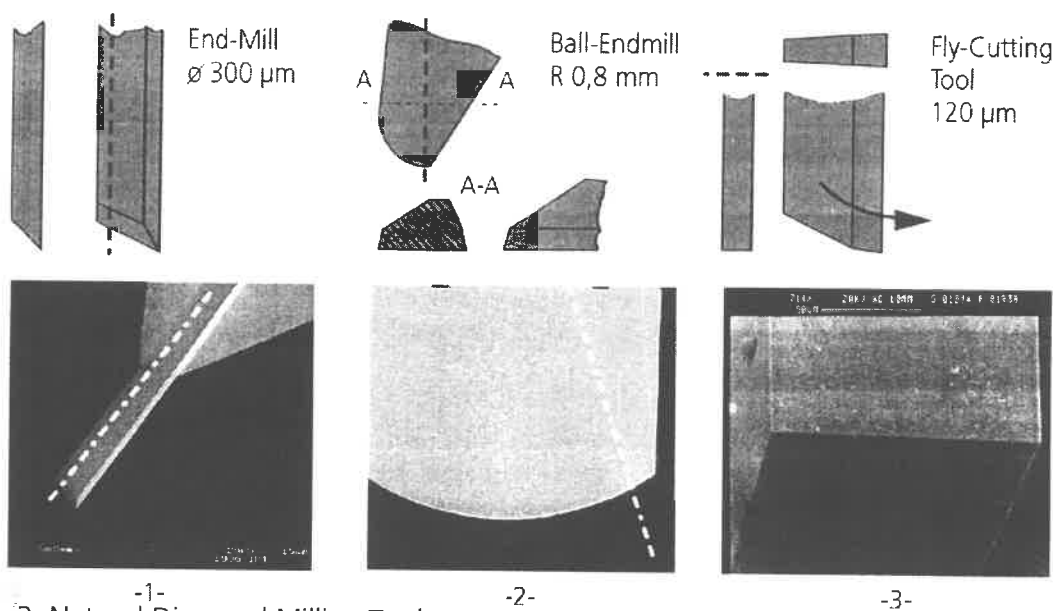


Fig. 3: Natural Diamond Milling Tools

Some applications for diamond milled surface structures are shown in fig. 4. Microstructured optics are used as a laser optic with spheric elements (1) or as a mold for reflective foils (2). The spheric elements were produced with a ball-endmill, the pyramid structure has been machined using a fly-cutting tool with a v-shaped diamond cutting edge. Other fields of applications are molds for fiberoptic elements (3,4), where small rib structures with optical surface quality and shape accuracies in the submicron range are needed. The pictured molds were fabricated with a diamond end-mill. The use of high speed spindle reduces the machining time, thereby increasing the economic efficiency of this technique. Interesting new products are also created in field of microfluidic systems. Here, surfaces can be structured in such a way as to increase, for example, heat transfer in biochemical and chemical catalytic converters. Other applications are micro-reactors to mix smallest amounts of reactive fluidics (5). The structures were also milled using an micro-endmill - minimum rib sizes of 1,5 μm width and 200 μm height have been machined into brass (6).

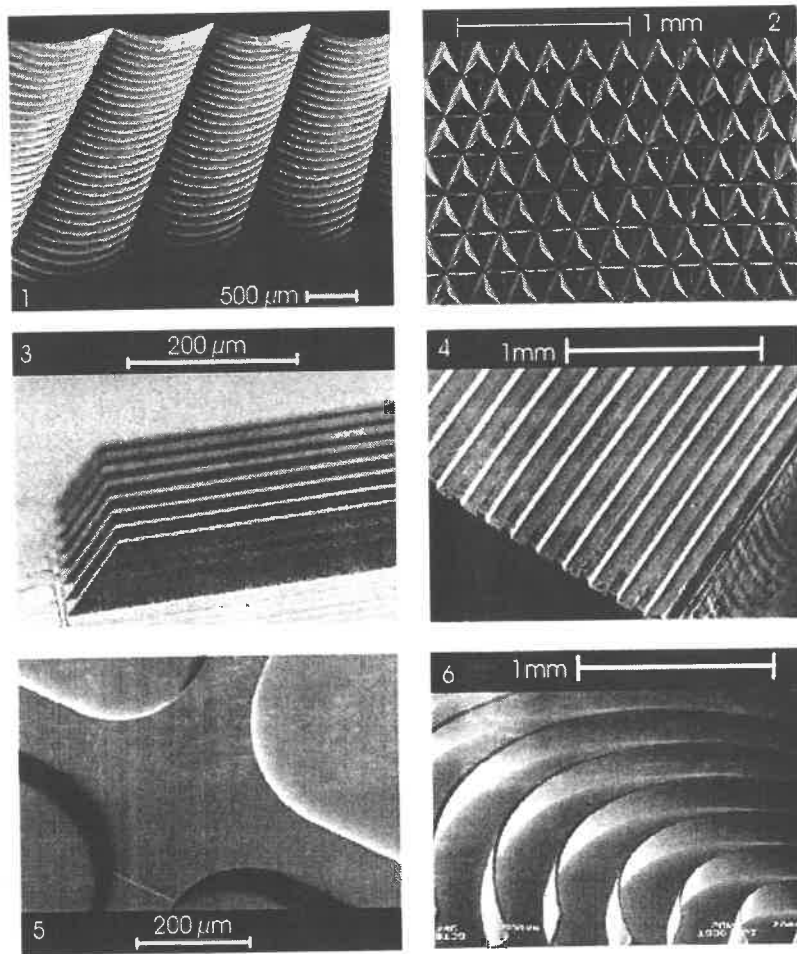


Fig.4: Workpieces produced by ultraprecision diamond milling

It emerged, even in very first tests conducted on an ultraprecision machine, that workpieces can be produced with shape accuracy, structure size and surface roughness far below previously known values by diamond milling. This conventional technique is well suited for the manufacture of prototypes and small series. However, the application with the highest economic potential is the fabrication of high precision molds and dies. Innovative products can be mass-produced cheaply when reproductive techniques such as galvanic forming in combination with injection molding or embossing processes are applied.

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Chip Dynamics in Diamond Turning

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Key Words: Chip dynamics, diamond turning

Introduction

In recent years, university and industrial research has provided a wealth of information concerning the material deformation processes involved in diamond turning. The focus of this work has been to predict the stresses and deformation in the region near the cutting zone to calculate the resulting forces and surface finish. One aspect that has not been studied is the dynamics of the chips as they leave the workpiece. This motion is an important issue because preventing the chips from touching the finished surface is crucial to achieving the optical quality surface possible with diamond turning. Therefore, a knowledge of the natural chip motion is the first step in designing a system for chip collection to prevent chip/surface interaction.

Chip formation has been shown [1] to be a highly plastic process involving severe local deformation of the workpiece material and dependent on the workpiece material properties, the friction between the chip and tool, and the geometry of the tool and workpiece. Therefore, the motion of the chip must be influenced by these generating forces and the chips, in turn, can play a role in the material flow process. Although the continuous chips created in most diamond turning operations have a small mass and a low stiffness, they remain connected to the root of the chip and can transfer forces back to the cutting zone. In addition, external forces can act on the chip after it leaves the workpiece and significantly influence the motion. External forces may be generated by gravity or by contact between the chip and one or more of the following: the workpiece, the tool, an air/oil stream or another section of the chip.

A study of chip dynamics has been undertaken to develop a model of the dynamics of diamond turned metal chips as they move away from the cutting zone and the workpiece surface. A 1997 ASPE poster paper [2] described machining experiments to study the effect of depth of cut, cutting speed, tool nose radius and tool wear upon chip motion in diamond turning. Work also detailed the importance of workpiece material as it was shown that lead, which does not strain harden, did not produce curled chips. Tests in strain hardening 6061-T6 aluminum and plated copper, however, both generated curled chips. The work presented here continues this effort and reports on experiments to show the influence of external forces upon chip geometry and motion.

Chip/Workpiece Interaction

Figure 1 shows the formation of a 6061-T6 aluminum chip as it contacts both the tool and the workpiece. The chip was generated with a 25 μm depth of cut at a cutting speed of 0.04 m/sec by plunging a 3.1 mm nose radius tool with a 0° rake angle directly into the face of a flat workpiece at a constant x-axis position. The infeed rate for each test was varied to move the tool to the desired depth of cut in less than 1/5 of a spindle rotation. This allowed the generation of a chip with a

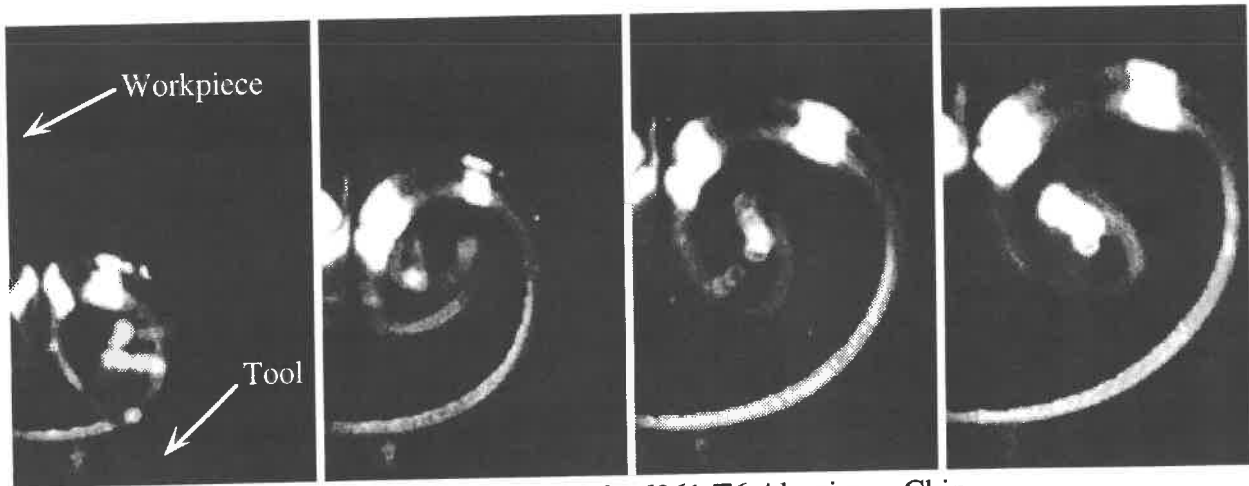


Figure 1: Formation of a 6061-T6 Aluminum Chip

constant cross-section over a majority of its length. As the chip is formed, it eventually contacts the workpiece surface, generating a bending moment at the root of the chip. This bending moment increases the radius of curvature of the chip. As the radius of curvature increases, the subsequent bending moment applied to the chip also increases. Thus, the radius of curvature of the chip increases along its length until it reaches some steady state value which is dependent upon the stiffness of the chip and the bending moment generated due to contact with the workpiece .

Effect of Gravity on Chip Deflection

The role of gravity upon chip motion was investigated by performing plunge cuts in plated copper with depth of cuts from 1 to 25 μm and cutting speeds from 0.15 to 7.65 m/sec. The first set of tests were performed with the tool in its normal orientation, facing upwards. The same cuts were then repeated with the tool facing downwards, thus reversing the direction of gravity during cutting.

Figure 2 compares radius of curvature data for the 5, 15 and 25 μm depths of cut when machining with the tool facing up and facing down. Each data point represents an average radius of curvature obtained for the chip generated during a plunge cut. Although differences in chip radius of curvature are small and inconsistent between the two tests, chip radius of curvature is generally larger when the tool is facing upwards. This phenomena is expected as gravity pulls the chip towards the tool face when the tool is facing upwards. The resulting bending moment causes the chip to straighten, creating a chip with a larger radius of curvature, as demonstrated in Figure 3 by a 25 μm chip generated at 0.74 m/sec. Conversely, gravity bends the chip away from the tool face when the tool is facing downward, causing the chip radius of curvature to decrease, as seen in Figure 4. Modeling the chip as a cantilever beam with a triangular cross section, a length twice its radius of curvature, and an end load corresponding to the weight of the entire chip generated during the plunge cut; the tip deflection for a 15 μm chip was calculated to be 73 μm . The difference in chip radius of curvature due to gravity, however, ranges from 100 to 500 μm . Thus, the influence of gravity in chip formation is believed to primarily influence the plastic deformation occurring in the shear zone and not the elastic deflections created along the length of the chip due to its weight.

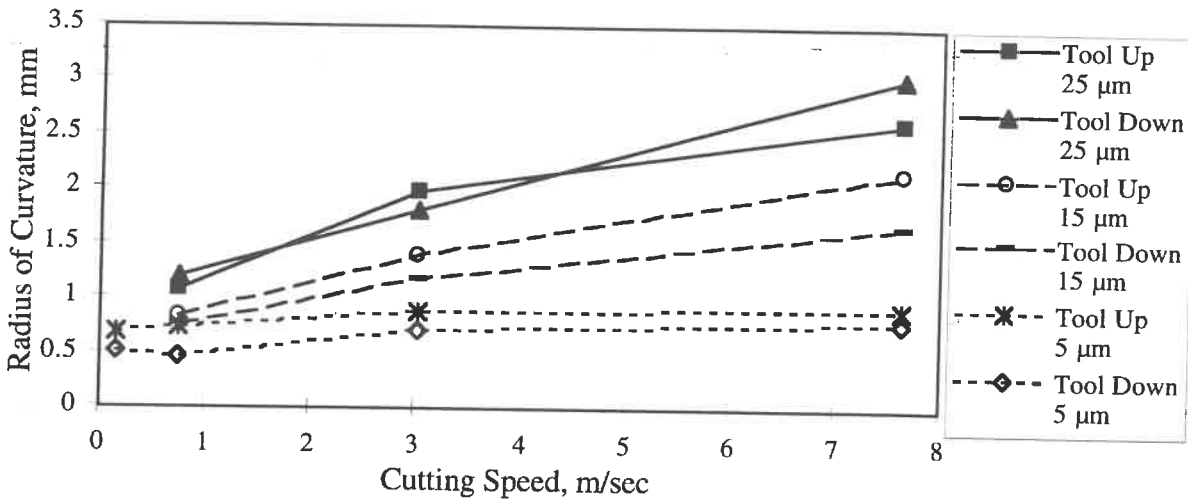


Figure 2: Change in Chip Radius of Curvature with Gravity



Figure 3: 25 μm Plated Copper Chip
(tool facing up)



Figure 4: 25 μm Plated Copper Chip
(tool facing down)

Influence of an Oil Stream on Chip Deformation

In many diamond turning operations, oil is utilized to steer chips away from the tool/workpiece interface. Thus, plunge cuts were conducted in plated copper to study the influence of an oil stream upon chip motion and geometry. Mobilmet Omicron cutting fluid was applied to the tool and workpiece surfaces through a 0.635 mm diameter nozzle with a 0.086 L/min flow rate. The first set of experiments involved a 5 μm depth of cut with cutting speeds from 1.46 to 1.56 m/sec, while the second set of tests were performed using a 10 μm depth of cut with cutting speeds from 0.73 to 0.77 m/sec. The first plunge cut in each set was performed without the oil stream and provided a standard of comparison for the other tests. Plunge cuts were then performed with the oil stream aimed from above the tool such that the oil contacted the tool perpendicular to its face. The oil stream was initially aimed at the tool/workpiece interface (Test Run #1). Additional cuts were then made while moving the oil stream along the center of the tool face, away from the tool/workpiece interface (Test Runs #2-4).

	Test Run	Chip Radius of Curvature (mm)
No Oil		0.596
Oil	1	1.003
	2	0.949
	3	1.129
	4	0.932

Table 1: Radius of Curvature Data
(5 μ m depth of cut)

	Test Run	Chip Radius of Curvature (mm)
No Oil		0.564
Oil	1	0.920
	2	1.244
	3	1.148

Table 2: Radius of Curvature Data
(10 μ m depth of cut)

Contact between the oil stream and chip generates a bending moment on the chip which will influence its motion and geometry. When the oil stream comes from above the tool face, the bending moment on the chip straightens the chip, increasing its radius of curvature. As the oil stream moves farther away from the tool/workpiece interface, the bending moment will increase, producing a straighter chip. Tables 1 and 2 present radius of curvature measurements for the 5 and 10 μ m depth of cuts obtained after performing the plunge cuts. An investigation of the data reveals that the oil stream increases chip radius of curvature. Although there does seem to be an increase in radius of curvature as the oil stream moves away from the workpiece, its influence is small and inconsistent.

Conclusions

Chip formation involves plastic deformation processes which can be influenced by external forces. Work has demonstrated that contact between the chip and the tool or workpiece influence its motion and resulting geometry. Work has also shown that the chip formation process is influenced by gravity, i.e. the weight of the chip, and that chip geometry can be changed by an oil stream. Such work points to the development of chip control systems which may be able to influence and alter the chip formation process.

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FABRICATION OF NON-IMAGING OPTICAL SURFACES

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INTRODUCTION

Optics for lighting systems are designed to collect and redirect all of the light from the source in the desired direction. However, there are a number of losses in the system. One source of losses, surface roughness, is the result of features produced by 1) vibrations between the tool and the workpiece, 2) the feed rate of the machining tool or 3) the response of the plastic to the cutting process. The goal of the experiments described here is to find the relationship between the diamond turning parameters and the surface finish of direct machined plastic surfaces as well as the optical performance of these surfaces.

SPECIMEN PREPARATION

Optical specimens were machined from blanks cut from a sheet of Lucite ES, medium strength extruded acrylic sheet. The fabrication of the optical specimens was performed using a Pneumo ASG2500 diamond turning machine. Two test parts were simultaneously machined at several machining conditions varying in both spindle r.p.m. and feedrate. The test conditions included spindle speeds of 100, 500, 1000 and 2000 rpm (surface speeds from 1 to 18 m/s) and feed rates of 0.5, 1, 2, 5, 10 and 20 mm/min. There were a total of 24 different cutting conditions studied with a total of 56 specimens produced (see Table 1). No liquid coolant was used during the machining of the test parts. A flow of air was directed at the tool/workpiece interface from below to blow away the chips. Since the cutting was intermittent, short chips were produced and there was no apparent buildup of chips on the tool or the workpiece. However, electrostatic forces tended to attract the chips to the workpiece.

The surface roughness of each sample was measured at the PEC using a Zygo NewView 100. Measurements were also made by J. Bennett at China Lake using a Talysurf profilometer with a 0.8 μm diamond stylus loaded onto the surface with 0.5 mg force. The measured surface roughness was compared to the theoretical finish that would be produced by a perfect diamond tool on a vibration-free machine tool. This surface finish will consist of a series of grooves at the spacing of the feed per revolution of the part. Larger feed rates increase the spacing of the grooves and thus the surface roughness while a larger tool radius decreases the curvature and thus the roughness. For a tool with a radius, R in mm, and a feed rate, f in mm/min, and a spindle speed, S_s in rpm, the calculated root-mean-square (RMS) surface roughness can be written as,

$$RMS(nm) = \frac{\left(\frac{f}{S_s}\right)^2}{26.8R} \times 10^6 \quad (1)$$

OPTICAL EVALUATION

Scattering measurements

A schematic of the test bed used to measure the optical performance of the diamond turned plastic optics is shown in Figure 1. This setup consists of a Hewlett Packard HeNe laser head serving as the light source, a rotation stage and an integrating sphere (with a photo detector) as shown. The diamond turned sample was placed between the integrating sphere and the laser source with the machined surface facing the integrating sphere. During the test, the sample was rotated to measure the intensity of the scattered light as a function of the angular displacement. The integrating sphere captures the scattered light while the direct beam passes through the sphere unobstructed. The

photodetector in the integrating sphere quantifies the amount of scattered light captured. The diameter of the inlet aperture on the integrating sphere is 19 mm and, combined with the 38 mm spacing between the sample and the entrance aperture, the sphere will capture all of the light scattered to a half angle of 14°. The exit aperture of the sphere is 6 mm and the half angle of the outlet beam is 0.85°.

Before optical testing of samples began, the apparatus was calibrated. Since only relative scattering intensities were measured, all measurements were related to a known minimum and maximum. The minimum is the condition where no scattering occurs due to the sample, therefore a regular data run was performed without a sample. Conversely, the maximum is where all of the laser's output is scattered into the sphere. This maximum was measured by reflecting a known percentage of the laser beam's intensity directly into the sphere.

Each sample was first oriented in the holder to find the zero incidence angle. This was done by moving the stepper motor until the beam reflected off the plano face of the sample was directed back onto the laser aperture. From this point, the stage was rotated -332 steps (-75°) and data was taken in 4-step (0.9°) increments through 664 steps (150°). The first sample measured was an uncut sample for use as a reference. Succeeding samples were then measured in the order that they were machined.

TIR Measurements

The distinguishing feature of Teledyne lenses is extensive use of TIR surfaces to direct light through them. A second set of optical measurements was made to determine the influence of the surface roughness on the ability of the TIR surfaces to reflect light. The setup is shown in Figure 2. The same laser source used in the scattering experiments was used to illuminate the sample. A glass roof prism was optically bonded to the sample, which, in turn, was temporarily attached to the entrance aperture of the integrating sphere by its edges. The light that was not internally reflected passed into the integrating sphere and was measured by the photodetector.

The integrating sphere was mounted on a circular scale with a pivot located so that, as the scale was rotated, the laser beam was always incident on the center of the hypotenuse of the prism.

The angles read off the scale, however, are not the angles of interest, so the corresponding angles of incidence on the machined surface must be calculated using Snell's law. If this is extrapolated over the several index changes shown in Figure 3, the Fresnel reflection losses may be calculated for an arbitrary set of interfaces with either circularly or diagonally polarized light using

$$\frac{E_{j+1}}{E_j} = \frac{n_{j+1}}{2n_j} \left(\frac{\cos r_j}{\cos i_j} \right) \left[\left(\frac{2n_j \cos i_j}{n_j \cos i_j + n_{j+1} \cos r_j} \right)^2 + \left(\frac{2n_j \cos i_j}{n_{j+1} \cos i_j + n_j \cos r_j} \right)^2 \right] \quad (2)$$

where E_j is the intensity of the beam in the j^{th} medium as shown in Figure 3. Fresnel reflection losses were thus

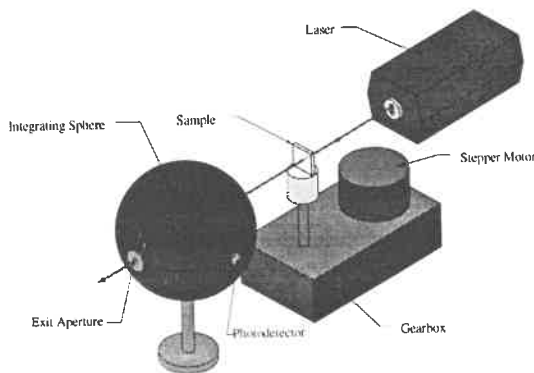


Figure 1: Setup of the integrating sphere used to measure the effectiveness of the diamond turned

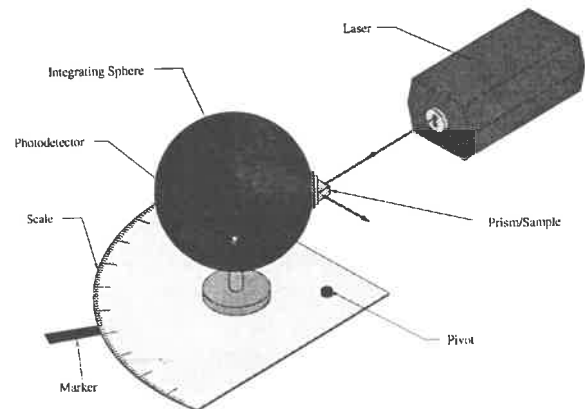


Figure 2 : Test setup for TIR efficiency measurements.

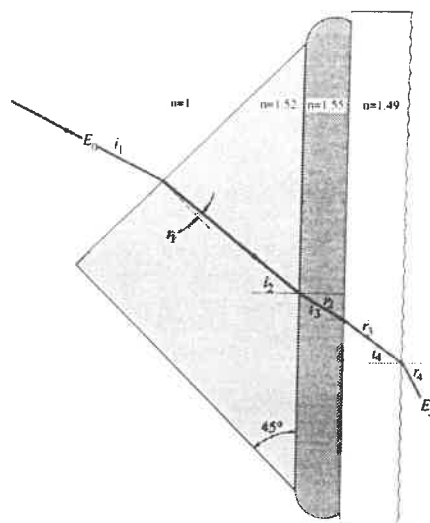


Figure 3: Light path for TIR experiments.

Test No.	Spindle Speed, rpm	Surface Speed, m/s	Feed rate, mm/min	Feed/rev, $\mu\text{m}/\text{rev}$	Calculated RMS, nm	Zygo data RMS, nm	Talysurf data RMS, nm	Relative Scattering	TIR critical angle, deg
1	100	0.93	20	200	981.93	1025	1043.0	0.126	46
2	100	0.93	10	100	245.48	303		0.092	
3	100	0.93	5	50	61.37	70		0.069	
4	100	0.93	2	20	9.82	14		0.024	
5	100	0.93	1	10	2.45	8	8.2	0.032	42
6	100	0.93	0.5	5	0.61	17		0.026	
7	100	0.93	20	200	981.93	1040		0.127	
8	500	4.66	20	40	39.28	42	39.3	0.061	42
9	500	4.66	10	20	9.82	12		0.024	42
10	500	4.66	5	10	2.45	5	6.0	0.018	42
11	500	4.66	2	4	0.39	6	6.5	0.027	
12	500	4.66	1	2	0.10	7	7.3	0.010	42
13	500	4.66	0.5	1	0.02	7		0.044	42
14	500	4.66	20	40	39.28	41		0.063	
15	1000	9.32	20	20	9.82	11	11.9	0.026	
16	1000	9.32	10	10	2.45	8	8.0	0.024	42
17	1000	9.32	5	5	0.61	9	8.8	0.032	
18	1000	9.32	2	2	0.10	10		0.059	
19	1000	9.32	1	1	0.02	7		0.054	
20	1000	9.32	0.5	0.5	0.01	7		0.044	
21	1000	9.32	20	20	9.82	14		0.023	
22	2000	18.64	20	10	2.45	*	1.3	0.021	42
23	2000	18.64	10	5	0.61	*	9.9	0.018	
24	2000	18.64	5	2.5	0.15	10		0.069	
25	2000	18.64	2	1	0.02	10		0.033	
26	2000	18.64	1	0.5	0.01	7		0.043	
27	2000	18.64	0.5	0.25	0.00	9		0.040	
28	2000	18.64	20	10	2.45	14	15.6	0.070	
Theory					0.00			0.000	42.15

Table 1: Test conditions and results. (* indicates specimens where the surface did not appear to be machined even though the programmed material removal should have been 50 μm . Bennett indicated that each of these surfaces were in fact machined and the surface finish is shown.)

calculated at each interface at varying angles to obtain theoretical data and normalize the light incident on the machined surface. Zero was set as the voltage reading for which no light entered the integrating sphere. The maximum was obtained by directly illuminating the sphere with the laser, taking the average of several voltage readings, and subtracting the fresnel reflection losses for the first three interfaces. All results presented here were normalized to these minima and maxima.

EXPERIMENTAL RESULTS

A summary of the experimental results are shown in Table 1. This Table shows the test number, the spindle speed, the surface speed at the mean radius of the workpiece, the feed rate of the x slide, the calculated rms surface roughness (from Equation (1)), the measured rms surface roughness from both interferometer and profilometer, the relative scattering intensity and the critical angle for total internal reflection. This angle compares favorably with the theoretical TIR critical angle of 42.15°. The experiments were performed in the order shown. Thus at each spindle speed the feed rate was reduced in steps from the maximum of 20 mm/min to the minimum of 0.5 mm/min followed by a repeat of the maximum feed rate. This repeat test was used to evaluate the process and the diamond to see if changes occurred as a result of the test sequence. The spindle speed was then raised to the next higher level and the test sequence was repeated. Two samples were machined for each test condition.

In general, the measured surface roughness is very close to the calculated value over the three orders of magnitude change in roughness. As would seem reasonable, the measured surface finish is always larger than the theoretical limit but very good surface finishes are possible using the diamond turning technique. To show that The lowest surface roughness was obtained at 500 rpm spindle speed. The reason for this optimum speed is not obvious to the researchers involved but may lie in the relationship between spindle vibration and rotation speed or some optimal cutting speed.

The measured surface roughness values produced during these experiments are in the range of commercial diamond turning facilities; that is, rms values are in the range of 8-30 nm depending on the material. Thus, the optical performance of the fabricated surfaces can be evaluated with the confidence that these surfaces are representative of current state-of-the-art in diamond turning practice.

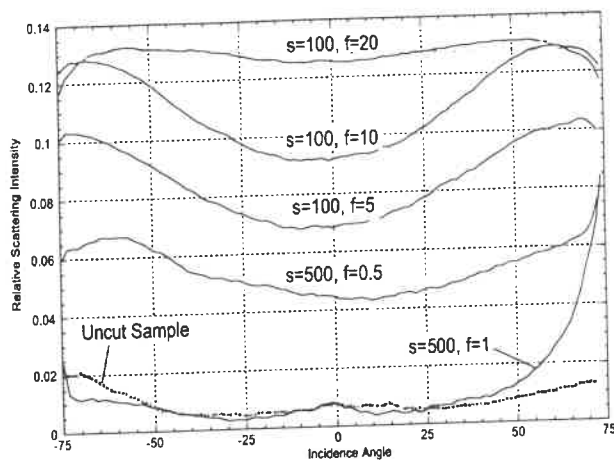


Figure 4: Scattering data for samples cut at 100 rpm.

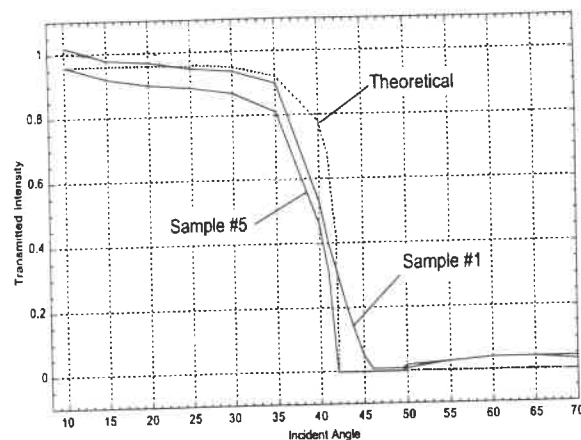


Figure 5: Theoretical and measured intensities for Sample #1. (Theoretical p-v roughness = 3298 nm, rms = 981 nm, max surface angle = 3.77°)

Relative light intensity scattering data from the test specimens for a few of the machining conditions enumerated in Table 1 is presented in Figure 4. Line plots of the normalized intensity of the scattered light versus angle of incidence from -75° to +75° are shown. At 100 rpm, reduction in the feed rate produced a surface with less scatter over the whole range of incidence angles. But as the feed gets less than 2 mm/min, little additional gain is observed.

It is apparent that, in general, a slower feed rate results in a surface that scatters less light. However, the specimens machined at 1000 and 2000 rpm were not better than those machined at 100 and 500 rpm. This is consistent with the measured RMS data which shows that the optimal surface finish was obtained for 500 rpm and 1 mm/min machining parameters. This surface also produced the lowest level of scattering. Thus the scattering data correlates well with the RMS finish data from the interferometer.

The Relative Scattering data presented in Table 1 is the average of mean scattering intensity from +5.4 to -5.4 degrees. Thus a single number can be used to represent a line of relative scattering as a function of incidence angle. These values can then be compared to the measured surface roughness of the samples. As would be expected, the very rough surfaces scatter relatively large amounts of light. The surfaces produced at 500 rpm had the lowest rms roughness (see Table 1), and also exhibit the smallest scattering, even better than surfaces machined at higher rpm with better theoretical surface finish. However, due to considerable variation in the data below 10 nm, strong conclusions could not be made about the reason for these apparently optimum machining conditions.

Measurements of the efficiency of total internal reflection in the plastic specimens are shown in Figure 5. The incident angle in the plot is calculated from the incident angle of the light onto the prism (this is the angle that is set for each experiment) and the change in that angle as the beam passes through the interfaces prior to the TIR surface. If the angle of incidence is less than the critical angle, most of the incident light is transmitted into the integrating sphere as shown on the left side of Figure 5. The dotted line is the theoretical transmitted intensity for perfect optical elements in the arrangement shown in Figure 3. As the light approaches the critical angle, the percentage reflected back into the prism (and therefore away from the integrating sphere) is increased and the transmitted intensity is reduced. At the critical angle, the transmitted intensity goes to zero.

The intensity of the transmitted light for the surface with the highest surface roughness (Sample #1) goes to a minimum much more slowly than the theoretical data. The reason for this lies in the variation in angle of incidence ($\pm 3.8^\circ$) across the part due to the features on the machined surface. Since the cusps on the machined surface subtend such a large angle in Sample #1, the incident light will be below the critical angle on one side of the groove while being above it on the other. The other samples tested all had significantly smaller grooves and the variation in the surface angle ranged from $\pm 0.75^\circ$ for Sample #8 in to $\pm 0.02^\circ$ for Sample #13. Because the measurements were taken in 1° steps around the critical angle, the resolution in the value of that angle is 1° . As a result, the critical angle for the remainder of the samples deviates little from the theoretical value.

CONCLUSIONS

The objective of this research was to determine the effects of the surface features created during diamond turning on the optical performance of plastic optics. To determine the role of surface finish, a number of acrylic test blanks were machined on one side with a diamond tool (1.52 mm nose radius) using different spindle speeds (from 100 to 2000 rpm) and feed rates (from 0.5 to 20 mm/min). These machining conditions produced a range of surface roughness from about 5 nm to 1 μm . The surface roughness of each specimen was measured using a white light interferometer (Zygo NewView) with a field of view of 0.13x0.18 mm and also with a contacting Talystep profilometer (0.8 μm radius at 0.5 mg load). The optical performance was determined by:

- 1) estimating the scattering and diffraction of light transmitted through the machined surface by passing a helium neon laser through the test blank and collecting the scattered light in an integrating sphere, and
- 2) assessing the efficiency of the total internal reflection by attaching a prism to the specimen, bouncing a laser beam off the back of the machined surface and measuring the light scattered from the front of that surface.

The results indicate that the measured roughness (using either the optical or contacting technique) of the surfaces produced on the diamond turning machine correlate well with the theoretical finish calculated from the grooves produced by the diamond tool. The lower limit of the achievable roughness on the PMMA material using the Pneumo ASG-2500 DTM at the PEC was about 5 nm rms. This limit was a result of one or more of the following potential causes: vibration of the machine structure, the geometry of the diamond tool or the details of the chip removal process. For these experiments with PMMA specimens, a spindle speed of 500 rpm (surface speed of about 5 m/s) was identified as producing the best surface finish. Spindle speeds larger and smaller than this value produced larger surface roughness for the same cross feed rate. The feed rate also has a significant effect on the finish, but below about 1 mm/min rate, no additional reduction in roughness was obtained for these experiments.

The results of the optical tests in comparison with the surface roughness measurement have revealed that, as expected, smoother surfaces perform better optically in a role where no scattering is desirable. Samples machined at 500 rpm consistently showed the best surfaces and optical performance characteristics. In fact a few samples even showed optical scattering performance characteristics comparable to those of the uncut surface shown in Figure 30. Comparing the relative scattering intensity from this figure with that shown in Figure 16 for the 500 rpm specimens shows no appreciable difference in scattering.

The TIR optical tests have revealed that TIR surfaces are relatively insensitive to surface finish. Even the roughest sample in the tests showed very little (less than 2%) transmission past the critical angle. There is, however, a possibility that some structure exists in the transmission curves that were missed by the relatively coarse resolution of the data acquisition. Future tests should be performed with automated equipment that can take data at less than 1° intervals. The prism could also be attached so that reflections off the exit face cannot enter the sphere in order to confirm the suspected origin of the 4% transmission rise past 50°.

Future tests will also utilize a tool with a smaller nose radius and investigate the influence of tool form errors on the surfaces produced and their optical performance. These experiments, proposed for 1998, will help to develop a general relationship between surface roughness and optical efficiency that can be used for designing new TIR systems.

PRECISION ENGINEERING AT POLAROID

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For nearly 30 years Polaroid has designed, built, and used a series of ultra-precision machines. We use them for three-dimensional profile measurements of optical surfaces on molding tools, and on molded plastic lenses and mirrors, and for precision grinding of optical surfaces in glass, semiconductors, and a variety of metals.

We routinely perform direct diamond cutting of semiconductors and of plastic and non-ferrous metal prototypes. We cut production masters for aspheric lenses and mirrors, Fresnel lenses and mirrors, and diffractive optical components for achromatization, athermalization, and beam division. Our applications in high-volume Polaroid camera products include six different aspheric optical parts without rotational symmetry.

We will describe these machines and many of the unusual optical systems that have been made possible through our use of them.

Friction-Based Design of Kinematic Couplings

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Abstract

Friction affects several aspects important to the design of kinematic couplings, but particularly the ability to reach its centered position is fundamental.¹ It becomes centered when all pairs of contacting surfaces are fully seated even though a small uncertainty may exist about the exact center where potential energy is minimum. For many applications, centering ability is a good indicator for optimizing the coupling design. Typically, the coupling design process has been largely heuristic based on a few guidelines [Slocum, 1992]. Several simple kinematic couplings (for example, a symmetric three-vee coupling) are compared for centering ability using closed-form equations. More general configurations lacking obvious symmetries are difficult to model in this way. A unique kinematic coupling for large interchangeable optics assemblies in the National Ignition Facility motivated the development of a computer program to optimize centering ability. However, space limits the description of the program to the basic algorithm. Currently the program is written in Mathcad™ Plus 6 and is available upon request.²

Background

Kinematic couplings serve many applications that require: 1) separation and repeatable engagement, and/or 2) minimum influence that an imprecise or unstable foundation has on the stability of a precision component. An object that is rigid, relatively speaking, requires six independent constraints to exactly constrain six rigid-body degrees of freedom. An object with one or more flexural degrees of freedom requires additional constraints; for example, a four-legged chair flexes torsionally to fit the shape of the floor. This paper deals only with six-constraint couplings supported through local surfaces and held in contact by a consistent nesting force. Quite often the nesting force is the weight of the object being supported, or it may result from a spring or

other force device. Ideally, the nesting force causes all surfaces to engage freely and with uniform loading.

Figure 1 shows the two classic types of kinematic couplings, the three-vee coupling (left) and the cone-vee-flat coupling (right). The symmetry of three vees offers several advantages: more uniform contact stresses, thermal expansion about a central point and reduced manufacturing costs due to identical features. Conversely, the cone (or the more kinematically correct tetrahedral socket) offers a natural pivot point for angular adjustments. The three-vee coupling is the natural choice for adjustments in six degrees of freedom.

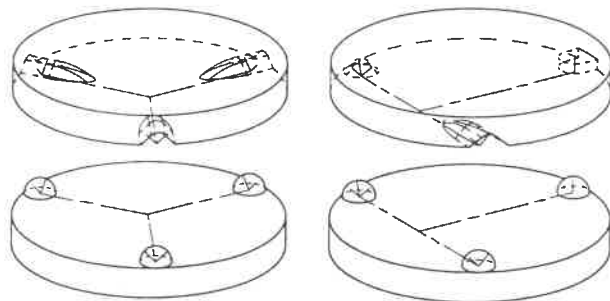


Figure 1 The three-vee coupling has six constraints arranged in three pairs. The cone-vee-and-flat coupling has six constraints arranged in a 3-2-1 configuration.

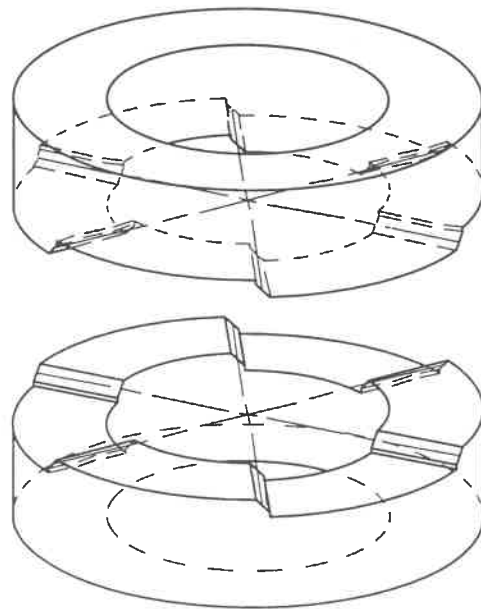


Figure 2 The three-tooth coupling forms three theoretical line contacts between cylindrical teeth on one half and flat teeth on the other half.

This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under contract No. W-7405-Eng-48.

¹ In this paper, the term kinematic coupling refers to any connection device based on pairs of contacting surfaces that provide six constraints in an ideal sense.

² Email requests to hale6@llnl.gov.

The local contact areas typical of these kinematic couplings are quite small and require a Hertzian analysis to ensure a robust design. Greater durability is achieved by curvature matching such as a ball against a concave surface and/or by using ceramic materials such as silicon nitride balls against tungsten carbide gothic arches. Designs based on line contact rather than point contact offer a significant increase in load capability. For example, a kinematic equivalent to three vees is a set of three balls in three conical sockets with either the balls or the sockets supported on radial-motion flexures. Alternatively, the three-tooth coupling shown in Figure 2 is based on three theoretical lines of contact formed between cylindrical and flat teeth. Each line constrains two degrees of freedom giving a total of six constraints. Manufactured with three identical cuts directly into each coupling half, the teeth must be straight along the lines of contact but other tolerances can be relatively loose.

Friction Effects

Friction affects at least four important characteristics of a kinematic coupling as indicated by order-of-magnitude estimates that all include the coefficient of friction μ .

- 1) Repeatability $\frac{f}{k} \approx \mu \left(\frac{2}{3R} \right)^{1/3} \left(\frac{P}{E} \right)^{2/3}$
- 2) Kinematic support $|f_t| \leq \mu f_n$
- 3) Stiffness $k_t = k_n \frac{2-2\nu}{2-\nu} \left(1 - \frac{f_t}{\mu f_n} \right)^{1/3}$
 $\approx 0.83 k_n$
- 4) Centering ability $\frac{f_c}{f_n} \approx 0.5 - 1.3 \mu$

Tangential friction forces at the contacting surfaces may vary in direction and magnitude depending how the coupling comes into engagement. This affects the repeatability of the coupling and the kinematic support of the precision component. The estimate for repeatability is the unreleased frictional force multiplied by the coupling's compliance. The estimate is derived as if the coupling's compliance in all directions is equal to a single Hertzian contact carrying a load P and having a relative radius R and elastic modulus E . The frictional force acts to hold the coupling off center in proportion to the compliance. This estimate will underestimate the repeatability if the structure of the coupling is relatively compliant compared to the contacting surfaces.

Kinematic support is important for stability of shape of the precision component. The estimate for

kinematic support simply gives a bound on the magnitude of friction force acting at any contact surface. A sensitivity analysis of the precision component will determine a tolerable level of friction that the coupling can have. This may drive the design to include flexure elements and/or procedures to release stored energy. If repeatable engagement is not so important, then constraints using rolling-element bearings offer very low friction. For example, a pair of cam followers that contact with crossed axes is equivalent to a ball on a flat but with twenty or so times less friction.

In some cases frictional overconstraint is valuable for increasing the overall system stiffness. Provided the tangential force is well below what would initiate sliding, the tangential stiffness of a Hertz contact is comparable to the normal stiffness [Johnson, 1985]. This was important for the National Ignition Facility where frictional overconstraint stiffened the first torsional mode of optics assemblies sufficiently to meet dynamic stability requirements.

Centering ability can be expressed as the ratio of centering force to nesting force and the estimate shown is typical. A larger ratio means the coupling is better at centering in the presents of friction. Later, it is convenient to express centering ability as the coefficient of friction where the ratio goes to zero. For the estimate, the limiting coefficient of friction is $0.5/1.3 = 0.38$. The coupling will center if the real coefficient of friction is less than the limiting value.

Centering Ability

The contacting surfaces of a kinematic coupling come into engagement sequentially unless it is placed precisely at the exact center. The path to center is constrained by the surfaces already in contact. For example, five surfaces in contact constrain the coupling to slide along a well-defined path. Four surfaces in contact allow motion over a two-dimensional surface of paths and so forth. Although there are infinitely many paths to center, only the limiting case is of practical interest for determining centering ability. Further, it is reasonable to expect the limiting case to be one of six possible paths that have five surfaces in contact.³ This point is demonstrated using the three-vee coupling.

For any given path to center, the centering force that results from the nesting force may be derived using Statics and the Coulomb law of friction. Figure 3 shows

³ The exception to this statement is the ball-cone constraint of the cone-vee-flat coupling since the cone provides only one constraint until the ball is fully seated. A tetrahedral socket remedies this situation.

examples of centering force (per unit nesting force) plotted versus the coefficient of friction. These curves were generated from closed-form equations yet to be discussed. Although the curves look simple, the equations are rather tedious to develop even when the coupling has simple geometry and the load is symmetrical. Compound this with the possible number of paths to center and it becomes obvious that a systematic, computer-based approach is essential for designing more general configurations of couplings.

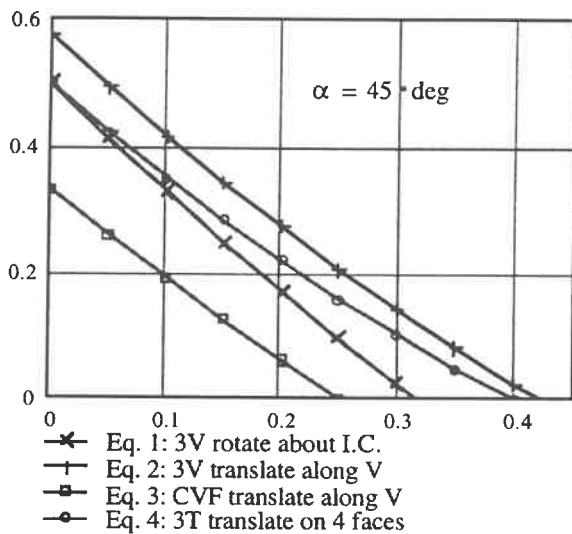


Figure 3 Centering force versus coefficient of friction.

Figure 4 shows a symmetric three-vee coupling rotating about its instant center to reach the center position. This path has five surfaces in contact and is the limiting case along with five other symmetrically identical paths. Equation 1 provides the centering force for this path assuming the nesting force is uniformly carried by three balls. Note, the sides of the vees are an angle α with respect to the plane of the three balls. In addition, there are two sets of symmetrically identical paths having four surfaces in contact. Equation 2 provides the centering force for the more limiting path where the coupling translates radially along one vee. The other path is rotation about one ball.

$$\frac{f_c}{f_n} = \frac{\sin \alpha - \mu \cos \alpha}{2(\cos \alpha + \mu \sin \alpha)} - \frac{\sqrt{3} \mu}{3 \cos \alpha} \quad (1)$$

$$\frac{f_c}{f_n} = \frac{\sqrt{3} \sqrt{4 + 3 \tan^2 \alpha} \sin \alpha - 4 \mu}{3 \left[\sqrt{4 + 3 \tan^2 \alpha} \cos \alpha + \sqrt{3} \mu \tan \alpha \right]} - \frac{\mu}{3 \cos \alpha} \quad (2)$$

This example shows that the path with five surfaces in contact has less centering force than either path with four surfaces in contact. This may not be universally true for a general kinematic coupling. That is, a path

with five surfaces in contact may have greater centering force than another path with four surfaces in contact. However as the coupling continues toward center, the centering force cannot increase as it picks up the fifth contact surface. Thus, we need only look at paths with five surfaces in contact to determine the limiting case.

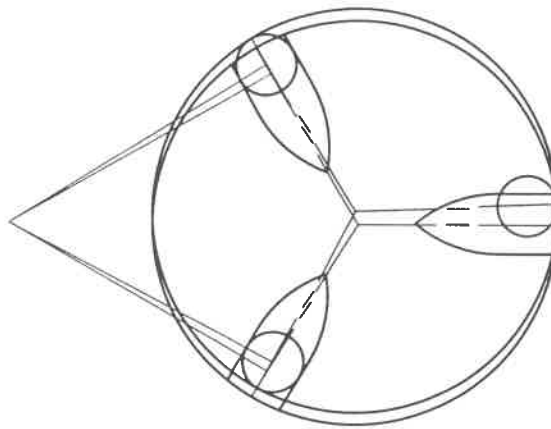


Figure 4 The limiting case for centering occurs when the three-vee coupling slides on five surfaces producing rotation about its instant center.

It is also useful to compare the centering forces for the other types of kinematic couplings. Figure 5 shows the cone-vee-flat coupling translating along its vee. This path is underdetermined for a conical socket but is representative of the limiting case. It was chosen to simplify the expression for centering force given in Equation 3. Referring back to Figure 3, it may come as a surprise that the cone-vee-flat coupling has the least centering ability of the three types. However, significantly improvement is possible by carrying more load with the cone and by increasing the cone angle.

$$\frac{f_c}{f_n} = \frac{\sin \alpha - \mu \cos \alpha}{3(\cos \alpha + \mu \sin \alpha)} - \frac{\mu}{3 \cos \alpha} - \frac{\mu}{3} \quad (3)$$

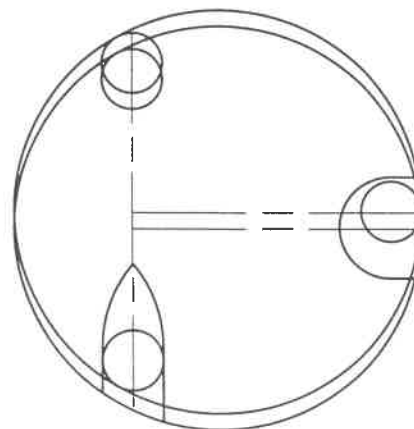


Figure 5 The limiting case for centering occurs when the cone-vee-flat coupling slides on the vee and flat with the cone seeking center.

Equation 4 gives the centering force for the three-tooth coupling as it translates on four surfaces. The centering force with five surfaces in contact is very difficult to model in closed form but behaves similarly to the limiting case for the three-vee coupling. For example, the limiting coefficient of friction for the three-tooth coupling is 0.319 at $\alpha = 45^\circ$ or 0.352 at $\alpha = 60^\circ$. The limiting coefficient of friction for the three-vee coupling is 0.317 at $\alpha = 45^\circ$ or 0.364 at $\alpha = 60^\circ$.

$$\frac{f_c}{f_n} = \frac{\sqrt{4 + \tan^2 \alpha} \sin \alpha - 4\mu}{2 \left[\sqrt{4 + \tan^2 \alpha} \cos \alpha + \mu \tan \alpha \right]} \quad (4)$$

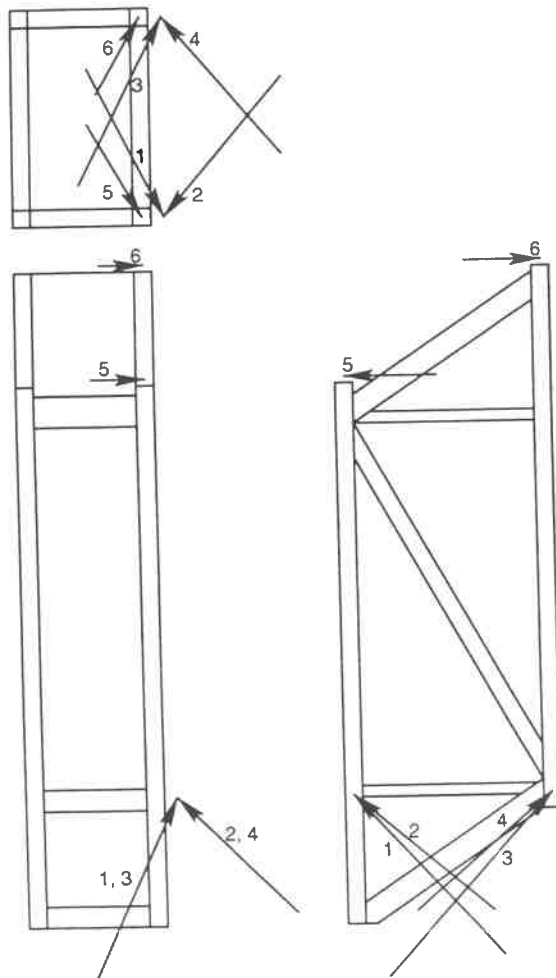


Figure 6 Three views of the optic assembly supported by six constraints. The arrows are proportional to forces.

Centering Optimization Algorithm

As discussed, the limiting coefficient of friction occurs along a path defined by five surfaces in contact. There are six such paths each corresponding to one of six surfaces not in contact. Using $[6 \times 6]$ transformation matrices, it is straight forward to reflect contact

stiffnesses to a global coordinate system (CS). Adding any five results-in a stiffness matrix for the coupling that has zero stiffness along the path. The eigenvector corresponding to the zero eigenvalue gives the direction that the coupling slides in the global CS. Using the same transformation matrices, a local sliding vector is determined for each surface. Then a force-moment vector is calculated for each surface using the Coulomb law of friction and a unit normal force. Transforming back to the global CS, the vectors are assembled into a matrix that when multiplied by a vector of contact forces gives the force-moment resultant on the coupling. The inverse of the matrix is useful because it gives the contact forces for a given coefficient of friction and applied load. Then the equation for the surface not in contact is solved for the coefficient of friction that makes its normal force zero. This is done for each surface not in contact, and the minimum is the limiting coefficient of friction.

Figure 6 shows one example of an optics assembly for the National Ignition Facility. This example has four angular parameters: the axis angle of two vee blocks (constraints 1-2 and 3-4); the more shallow face angle of the vee blocks (constraints 1-3); the steeper face angle (constraints 2-4); and the angle of the upper constraints. The user determines the optimum by adjusting the nominal parameter vector θ based on curves that show the effect of individually varying parameters. Figure 7 shows the optimal parameter set for this example.

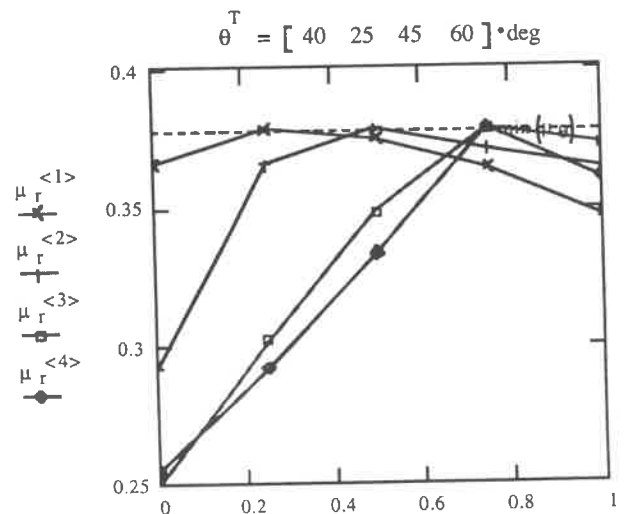


Figure 7 Limiting coefficient of friction versus varying model parameters. All curves pass through the nominal parameter set as indicated by the dashed line.

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SPATIAL FREQUENCY DOMAIN ERROR BUDGET

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1. Abstract

The aim of this paper is to describe a methodology for designing and characterizing machines used to manufacture or inspect parts with spatial-frequency-based specifications. At Lawrence Livermore National Laboratory, one of our responsibilities is to design or select the appropriate machine tools to produce advanced optical and weapons systems. Recently, many of the component tolerances for these systems have been specified in terms of the spatial frequency content of residual errors on the surface. We typically use an error budget as a sensitivity analysis tool to ensure that the parts manufactured by a machine will meet the specified component tolerances. Error budgets provide the formalism whereby we account for all sources of uncertainty in a process, and sum them to arrive at a net prediction of how "precisely" a manufactured component can meet a target specification. Using the error budget, we are able to minimize risk during initial stages by ensuring that the machine will produce components that meet specifications before the machine is actually built or purchased. However, the current error budgeting procedure provides no formal mechanism for designing machines that can produce parts with spatial-frequency-based specifications. The output from the current error budgeting procedure is a single number estimating the net worst case or RMS error on the work piece. This procedure has limited ability to differentiate between low spatial frequency form errors versus high frequency surface finish errors. Therefore the current error budgeting procedure can lead us to reject a machine that is adequate or accept a machine that is inadequate. This paper will describe a new error budgeting methodology to aid in the design and characterization of machines used to manufacture or inspect parts with spatial-frequency-based specifications. The output from this new procedure is the continuous spatial frequency content of errors that result on a machined part. If the machine does not meet specifications, the procedure identifies the sources of the critical errors. We would then evaluate these errors and either reduce the errors through design improvements or modifications to cutting parameters (spindle speed, feed, etc.) or select a different candidate machine if improvements were not practical.

2. Background

The principles of designing precision instruments for meeting challenging tolerance requirements have a rich history [1]. Likewise, the methodologies for analyzing the errors in experimental data and performing differential sensitivity analyses are well-documented [2,3]. Yet the first clear formalization of error budgeting applied to precision engineering appears to originate in the analysis by R. Donaldson during the design of the Large Optics Diamond Turning Machine at Lawrence Livermore National Laboratory (LLNL) [4]. Donaldson's formalism is referenced in current textbooks [5] and is the basis for subsequent machine designs at LLNL [6].

Figure 1 shows flowcharts for both the conventional and the new error budget procedure and how they differ. The upper portion of Figure 1 shows Donaldson's flowchart that illustrates the mapping of error sources onto a work piece geometry. The first step of the conventional error budget is to identify the physical influences that generate the dimensional errors that propagate through the machine tool. These include effects such as thermal gradients and temperature variability, bearing noise, fluid turbulence in cooling passages, way non-straightness, etc. The next step is to determine how this source couples to the machine. A coupling mechanism converts these physical influences into a displacement that has a direct influence on machine performance. An example of a coupling mechanism is the thermal expansion that may transform a time-varying heat source in the vicinity of the machine into a machine way distortion. These displacements represent dimensional changes in the system. A single peak-to-valley number is usually used to quantify the dimensional changes, not differentiating between the spatial frequency content of the error. The next step is to sum all the contributing errors using an appropriate combinatorial algorithm. Literature suggests a variety of combinatorial algorithms [7]. The last step in the error budgeting procedure is to transform these errors into the work piece coordinate system. To convert these machine displacements into the errors that would reside on the work piece surface in the directions of interest, we must consider the tool path (i.e. feed rates, spindle speeds, etc.). The output from this procedure is a single number predicting the net error that would result on a machined work piece. We would then compare this number to the part specifications. If the prediction meets target specifications, we would accept the machine design under evaluation. If the prediction does not meet specifications, we would evaluate methods to improve this design by observing which

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sources are the dominating contributing errors. In this way we can evaluate the cost versus accuracy of different candidate designs. If improvements can be made to an existing design, we would make those changes to the error budget and reevaluate the net error. If the modifications are not practical, we would then consider an entirely new design, or possibly reevaluate the specifications.

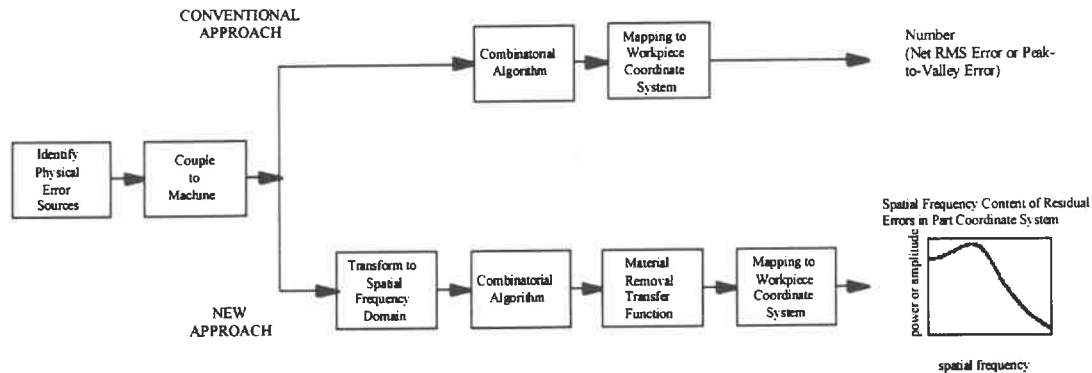


Figure 1. Flowchart comparison between conventional approach and new approach.

3. Technical Approach

The lower portion of Figure 1 shows the new error budget approach. The first two steps, identifying the sources and how they couple to the machine, are identical. However, the next step in the new approach converts the elemental errors into the frequency domain. The next step is to combine the errors in the frequency domain. The combinatorial rule is a completely new algorithm with a statistical foundation. The next step considers the machining process. The conventional approach assumes that the machine and cutting process don't damp or amplify the error sources, while the proposed approach considers the dynamics of the cutting process. The last step is to transform the errors into the part coordinate system. The output from this process is the continuous spectrum of errors that would result at all spatial frequencies on the part.

3.1 Combinatorial Rule

We have developed a combinatorial rule for the addition of the frequency content of each elemental error. The key to the combinatorial algorithm is to consider the spectrum of each elemental error as the sum of sinusoidal errors at specific frequencies. The addition of two sinusoidal signals at a given frequency results in a sinusoidal signal with the same frequency, but the amplitude can vary anywhere from the direct difference to the sum of the two amplitudes depending on the phase shift between the two signals. We first identify all elemental errors that are correlated and appropriately sum the amplitudes of these errors. We then consider the phase shift between the remaining elemental errors to be uniformly distributed variables between 0 and 2π . We have analytically shown that the expected value of the square of the net amplitude is equal to the sum of the squares of the amplitudes of each elemental error. This is equivalent to saying that the expected net power spectral density (PSD) is the sum of the elemental PSDs. Furthermore, we can now determine the probability distribution function of the net error with the use of a Monte Carlo simulation. The 95% confidence limit of the net PSD is approximately three times the mean, and the 99% confidence limit is approximately 4.6 times the mean. This is significantly less than the worst case error. For example, if 25 errors of equal amplitude were summed, the worst case net PSD would be over eight times larger than the 95% confidence limit, and over five times larger than the 99% confidence limit.

3.2 Cutting Process Transfer Function

The purpose of the cutting process transfer function is to convert the motion of the tool in free space to the motion of the tool in the part during the cutting process. This step is necessary because current error characterization procedures measure the error motion of the tool in an open loop sense. The loop is closed when tool is in contact with the part during the cutting process. Differences occur when the loop is closed due to static and dynamic stiffness of the machining system. The conventional error budgeting procedure assumed that the measured motion of the tool in free space is the same as the motion of the tool in the part during cutting, or in other words it assumes that the transfer function equals one. This may be true for a specific frequency band, although we assume that the cutting process will act as a filter with the ability to amplify or suppress the input error motion. For example, if the frequency content of one error were close to the machine resonance, we would expect this error to be amplified. In general, however, during precision machining we (1) design machines with high first modes, and (2) avoid machining conditions that are at the machine resonances. For precision operations, we expect that the machining

operation will appear as a low-pass filter, where the low frequency gain is one and the high frequency gain decreases.

3.3 Mapping the Errors into the Work piece Coordinate System

Given the frequency content of the error motion of the tool during cutting, we must take into consideration the path of the tool and the tool geometry determine the frequency content of the residual surface errors on the work piece. Typical tools with a round cutting edge impart a nominal surface finish, or scalloping, during turning, even for a process with no errors. Next, we consider the exact path of the tool during the entire cutting procedure to map these errors onto the relevant work piece coordinate system.

For example, during a facing operation on a diamond turning machine, the part turns while the tool remains stationary. Consider the spatial frequency content of a radial trace across the work piece. The turning process can be considered a sampling mechanism. The radial trace is composed of the time domain sampling of the tool motion once every revolution of the part. Once every revolution, the tool falls on the radial trace of interest, leaving behind the signature of the tool as well as any error motions.

The description of the process so far has been in the time domain. However, we are interested in the frequency domain. Sampling in the time domain can be decomposed into a multiplication procedure of the original time domain signal by a series of impulses. Since multiplication in the frequency domain is equivalent to convolution in the frequency domain, the sampling procedure is converted to the frequency domain by a convolution process. Note that unavoidable aliasing occurs for errors with higher frequency content than the rotational speed of the spindle. Note also that errors at frequencies that are an even multiple of the spindle speed (such as 'synchronous' spindle errors) do not appear on the radial trace due to this aliasing.

The imparting of the tool geometry onto the work piece can be considered a convolution in the time domain. Conveniently, convolution in the time domain is equivalent to multiplication in the frequency domain. Therefore, the imparting of the tool geometry onto the work piece in the frequency domain can be considered a filter.

4. Validation through Experimentation

We are in the process of validating this procedure through actual machining tests. Our test bed is a T-based lathe. We will demonstrate this procedure for facing and cylindrical turning of copper using a diamond tool. For these simple cases, the dominant elemental errors include the feed axis straightness and the spindle axial, radial and tilt errors. During an actual error budgeting procedure, the form for the elemental error spectrums must be assumed based on previous experimentation and experience along with expert knowledge and analysis of the machine design. However, for this validation we will actually measure the frequency content of the elemental errors and then employ this procedure to predict the net spectrum of errors that will reside on our test parts. We will then machine these test parts and characterize the errors in the frequency domain using a combination of surface finish and form measuring instruments. The predicted spatial-frequency spectrum will be compared to the measured errors for final validation.

4.1 Results from Spindle Tests

The test machine has an air-bearing spindle with a DC brushless motor, pulse-width-modulated controller and a resolver for feedback. We measured axial and radial error motions of the spindle using a test mandrel with two precision balls and capacitance gages. Because we are interested in the spatial rather than temporal frequency content of the errors, we used an encoder to trigger data acquisition, gathering one thousand points every revolution of the spindle. We recorded twenty revolutions of data for spindle speeds from 60 RPM to 1500 RPM in steps of 60 RPM. As expected, the air-bearing spindle is very repeatable with sub-micrometer levels of asynchronous motion. However, the error characteristics drastically change at different spindle speeds. For example, the axial motion at 840 RPM spindle speed has a synchronous error with a dominant lobing of 17, 18 and 19 cycles per revolution as shown in

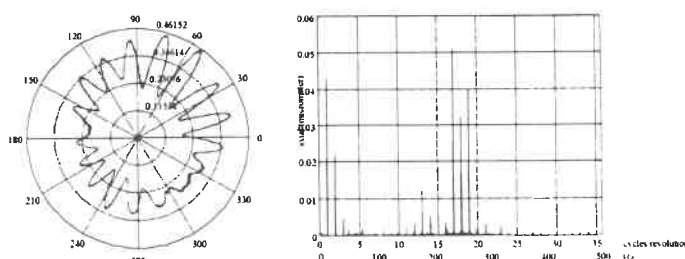


Figure 2: Axial Errors at 840 RPM

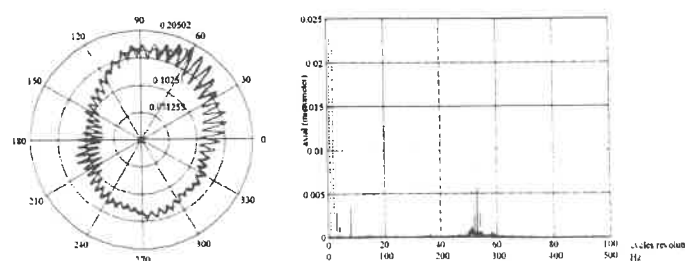


Figure 3: Axial Errors at 300 RPM

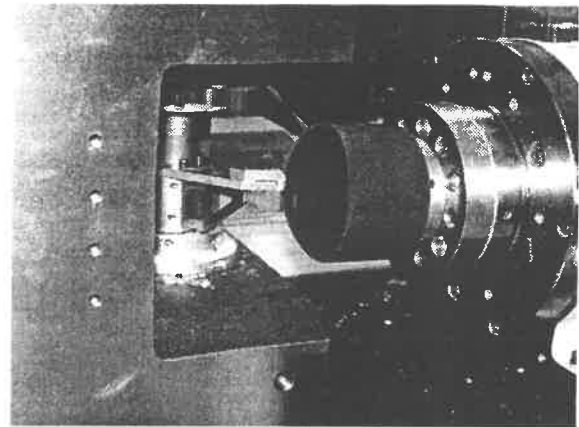
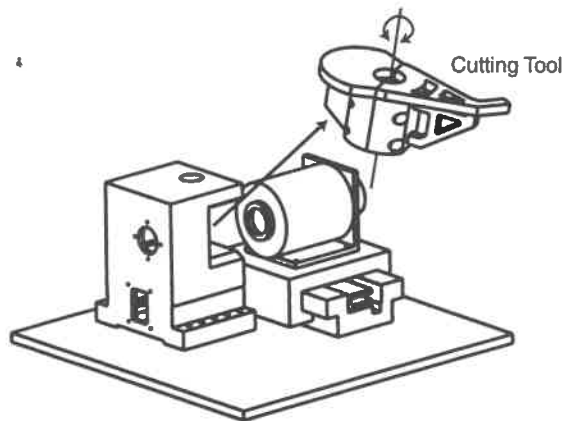


Figure 1: Schematic drawing of the prototype turning machine and a close-up photograph of cutting tool and a part.

and has a usable travel of 14 cm. The cross slide carries a Professional Instruments 10,000 RPM air bearing spindle with an internal brushless DC motor and a 10,000 count/rev encoder. Both the spindle and cross slide are conventional axes which we have overdesigned for this application in order to concentrate efforts on the development of the fast tool servo axis.

The rotary fast tool servo axis consists of a motor, coupling, shaft, bearing assembly, toolholding arm, and encoder. The toolholding arm is an 8 cm long bolted structure with a 9 mm diameter tungsten carbide cutting tool mounted at its end. This is about the shortest length that can still cut all of the lens surfaces which may be prescribed. The arm assembly bolts to a 25.4 mm diameter shaft that rotates in two back-to-back sets of angular contact ball bearings. The bearings have an ABEC-9 accuracy rating ($2.5 \mu\text{m}$ error motions), and are protected by a set of labyrinth seals. The grease in these seals appears to provide beneficial damping to the axis, but we have not yet quantified the effect. The actuator for the tool axis is a brushless servomotor rated at 2330 W and 9.6 Nm continuous torque output². This is standard product chosen based on its high ratio of torque output to rotary inertia, and its relatively low inductance.[3] A flexible coupling connects the toolholding shaft to the drive motor, but in the next design iteration we expect to eliminate the coupling in favor of a frameless motor mounted directly to the shaft. This will result in a lower cost, higher bandwidth system. The position sensor for this axis is a 15,430 count rotary encoder connected to a digital signal processing board with a 12-bit A/D converter.³ The electronics interpolate the sinusoidal waveforms to increase the resolution to 63,201,280 counts/rev. The result is a sensor that achieves about $0.1 \mu\text{rad}$ angular resolution, or an equivalent 8 nm linear resolution at the tool tip.

A digital signal processing board residing in a host 486-based desktop computer implements the control algorithms. The DSP board contains a Texas Instruments TMS320C31 floating point processor and four *Industry Pack* sites which hold a 12-bit D/A converter, digital I/O for communicating with the tool axis sensor system, and a quadrature decoder card for reading the cross slide and spindle axis encoders. The DSP performs all real time tasks at a 10 kHz control-update frequency while the host computer provides monitoring and user interface functions.

System Modeling and Controller Design

A simple but useful model of the fast tool servo axis consists of two masses inter-connected by a spring and dashpot. The two masses are the motor rotor and the toolholding shaft, and the spring/dashpot models the compliance in the coupling between them. The small amount of damping is provided by bearing friction and the grease-filled labyrinth seals. The primary structural mode of this system is located at 920 Hz, and the second mode (not captured by the simple model) has been experimentally measured at 2250 Hz. The low

²Aerotech model BM-1400, Aerotech, Inc., Pittsburgh, PA

³MicroE Inc., Natick, MA.

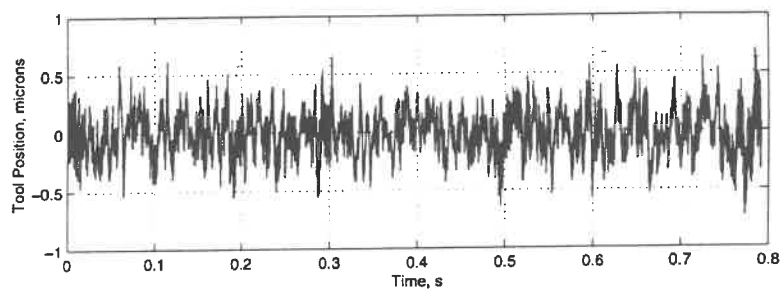


Figure 2: Steady state position error of the closed loop fast tool servo axis.

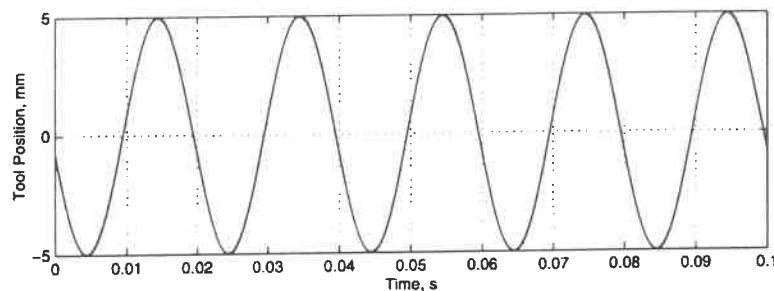


Figure 3: Example of a 1 cm peak-to-peak trajectory followed at 50 Hz.

frequency behavior of the plant is amplitude dependent and influenced by static friction and motor cogging effects. We do not have a well-validated model for this behavior yet, but have designed the controller with a high enough bandwidth that we can ignore this characteristic at the present time.

The primary controller for the fast tool servo axis is quite straightforward. We use lag compensation to increase low frequency stiffness, and double lead compensation to achieve 50 degrees of phase margin at a crossover frequency of 200 Hz. A 3rd-order digital Butterworth notch filter attenuates the 920 Hz component from the control effort to lessen the excitation of the primary structural resonance. The crossover frequency gives a rough measure of the bandwidth of the closed loop system, but experimental traces later in this paper show that the fast tool servo is capable of following sinusoidal trajectories at frequencies of up to 500 Hz with reduced amplitudes. In ongoing work, we are studying the application of repetitive control [5] and adaptive feedforward control [4] for reducing servo following errors in actual turned parts.

Results

The performance specifications for the prototype machine require that we turn test parts to $1\text{ }\mu\text{m}$ form accuracy. We have not yet achieved that goal, but can demonstrate progress towards it. Figure 2 displays the steady-state controlled position gathered at 10 kHz. The root mean square position is $2.44\text{ }\mu\text{rad}$ which corresponds to 195 nm RMS at the tool tip. A closer look at the position error shows that it is almost entirely composed of components at the structural resonant frequencies of 920 Hz and 2250 Hz. We expect this level of position resolution to be adequate provided that it does not lead to an unacceptable surface finish in the final parts.

The other main test for the fast tool servo axis is its ability to follow high acceleration sinusoidal trajectories. Figure 3 shows a 125 mrad peak-to-peak sinusoidal trajectory at 50 Hz. This corresponds to 1 cm peak-to-peak at the tool tip, and an effective tool tip linear acceleration of 490 m/s^2 , or 50 g's. The tracking error between the reference and followed trajectory is dominated by phase shift which we assume can be compensated for with more careful feedforward and repetitive control. Performing a least-squares fit on the position data and subtracting the best sinusoid leaves us with a residual RMS error of $12\text{ }\mu\text{m}$. The fast tool servo axis is also required to follow much higher frequency trajectories at lower amplitudes. These are typically harmonics of the dominant underlying frequency which serve to "sharpen" the trajectory. Figure

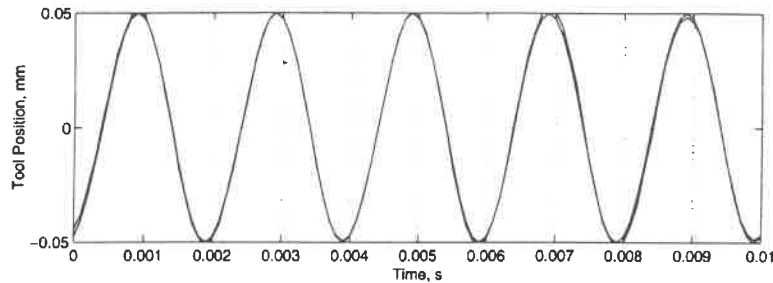


Figure 4: Example of a 100 micron peak-to-peak trajectory followed at 500 Hz.

4 shows an example of the fast tool servo following a 1.25 mrad peak-to-peak trajectory at 500 Hz. This corresponds to 100 μm at the tool tip, and again a peak acceleration of 50 g's. Again, removing the best-fit sine wave in this case compensates for the phase-induced error and yields a residual RMS following error of 1.7 μm .

Conclusions and Future Work

In this paper we have presented the concept of a rotary fast tool servo, and demonstrated that this topology allows for a relatively simple mechanical system with exceptional acceleration capabilities. We are currently investigating repetitive control algorithms that will allow us to reduce tracking errors to sub-micron levels under actual cutting loads. In addition, we will be developing calibration procedures for this topology, and confirming form accuracy and surface finish on actual parts. We also expect that achieving acceptable accuracy will require a mechanically stiffer arm design with a dedicated tool height adjustment mechanism.

Acknowledgments

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PLANARIZATION OF BGA DEVICES PRIOR TO ELECTRONIC TESTING

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Abstract

Planarizing BGA chips to test for reliability and electrical performance could greatly simplify the world of testing. The mechanical requirements of a high-performance production contactor include a high-integrity, repeatable connection between the package and the test board (1). The advantage of planarization is that it minimizes the variation in height of the bumps, reducing the problems in the testing phase and perhaps in the assembly phase as well. It also opens the door to designing simpler contactors to improve the electrical performance at test. Considering semiconductor devices are 100% tested to ensure standards of reliability and quality are maintained, the temporal contact made with the chip during test must be reliable as well as robust. Traditionally, flexible contacts like the bent leads and spiked compressible pins (pogo pins) were used to test Ball Grid Array (BGA) devices having pitches of 1.27mm or greater. Because it is very difficult to form a perfectly planar array of contacts on the tester or the device to be tested, the need for mechanical flexibility has persisted. As the industry moves to tighter pitches in the array, there is an increasing challenge to modify the testing methods. In order to insure repeatable contact force, less variation in ball heights or more planarity is required.

It is proposed that by planarizing the solder balls of the BGA chip to a micron level of flatness, less deflection is necessary and more rigid contactors can be developed. Some mass planarization attempts to address the issues of ball height variation are being made, however this is not being done at the micron level. Currently, we are investigating the planarization of BGAs and μ BGAs™ using diamond machining. Planarity ranging from 2-12 μ m has been demonstrated. As a result, a long-term goal is to introduce a system for planarizing the solder ball tops of a μ BGA™ prior to test. A diamond flycutting machine can be added to the manufacturing process provided it is a smooth and uninterrupted transition.

Improved planarity allows for the use of rigid contactors, which is characterized by Hertzian contact principles when contacting the device under test (DUT). The functional requirements of this design are to break through the oxide layer and enable sufficient mating to establish good electrical contact. The BGA array can mate with modest applied pressure to the planarized array of rigid mechanical contacts with approximately shaped tips to cause predictable Hertzian contact deformations to penetrate the oxide layer. This will result in a more robust and repeatable high-speed electrical testing method at an overall lower cost. Furthermore, this method provides the means for BGAs and subsequently μ BGAs™ with tighter pitches and tolerances to revolutionize testing in the industry.

Procedure

One of the initial steps preempting the planarization investigation was a statistical analysis on the existing solder bump height variation. Each chip had a 5 x 8 array of solder bumps uniformly spaced as shown in the Figure 1.

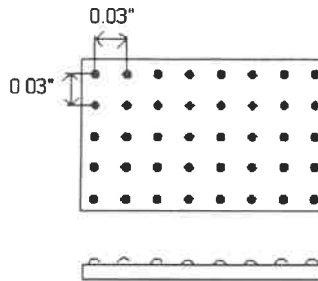


Figure 1: Ball Grid Array (BGA) Chip

With a pitch of 0.03" (0.76 mm), this chip can be categorized as a μ BGA™. BGAs and μ BGAs™ have become increasingly popular because designers have the ability to pack a high number of I/O's into a smaller area. In addition to this, the solder balls on BGAs and μ BGAs™ self-align when mated to the PC Board (2). With the larger pitched devices, solder ball height variations are not as much of a concern. However this is not the case as the devices become smaller. The task of testing suggests that new methods must be established for the smaller pitched devices. Therefore, we first determined the solder ball height variation of each of the bumps using a Coordinate Measuring Machine (CMM). The CMM was programmed to index a 1mm-diameter probe along each row of solder bumps at 0.03" (.762mm) increments. At each increment, the probe records the height of the solder ball at that point. The probe applies approximately 0.8 grams of pressure at contact.

A total of twelve chips were measured on the CMM. Linear regression analysis was performed on the heights to determine the best-fit plane through peaks of the solder balls. The co-planarity values for each device were normalized between the actual heights and the best-fit height. We defined one co-planarity value per device measured as the maximum difference between the best-fit plane and the actual data. Prior to planarization, the co-planarity values ranged from 30 to 40 μ m. After planarizing the tops of the solder balls, the heights would be re-measured in the same fashion.

Diamond Machining

After measuring the height of the solder balls on each chip, the tops were all machined down into a flat plane parallel to the back of the chip. Diamond machining the solder balls was chosen as the tool for planarization because of the excellent surface finish and the attainable precision. There was some compliance in the substrate of the chip, which caused concern when planarizing them. The technician expressed some hesitation in doing this because he felt that compliance would cause some of the solder balls to tear off, therefore adversely effecting the machining process. Machining them was nevertheless attempted, and proved to be successful. The diamond turning machine used was a 5-nanometer resolution machine. Although it was a dry cutting process, it exhibited slight sticking to the tool. The radius of the diamond tool used was 0.030". These devices were mounted to a fixture with double-sided adhesive and all machined at the same time. Approximately a quarter of the nominal chip height was removed.

The first few BGAs and μ BGAs™ were used to set-up the procedure and converge on machine parameters such as feed rate, depth of cut and spindle speed. The first chip was initially machined with a feed rate of 0.0025"/sec, depth of cut of 0.002", and a spindle speed of 1200rpm. The initial choice of parameters produced poor surface finish. This is shown in *Figure 2*. The difficulty in cutting solder lies in the fact that it is a soft material. However the changes in feed rate and depth of cut must have helped moderately because the surface finish showed considerable improvement as we tweaked the cutting parameters. The resulting surface finish after the changes were made produced an almost mirror-like surface as shown in *Figure 3*.



Figure 2: Surface finish of a diamond turned solder ball using initial parameters

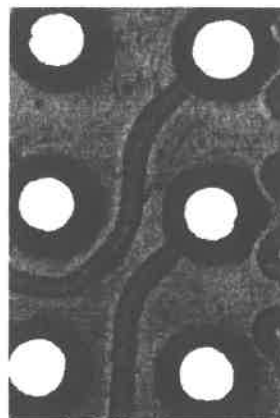


Figure 3: Surface finish of an array of diamond turned solder balls after parameter adjustments

Figure 2 also shows burring and buildup of solder after planarization. This again is the result of machining "soft solder". Planarizing BGAs with soft solder causes buildup on the tool. When comparing *Figures 2 and 3*, it can be seen that the diamond machining had to be adapted to the softness of the solder. It is worth noting that the technician noticed that a small oxide layer had formed on the diamond-machined surface as well after several days. It was looked at under a microscope just after machining and there was no noticeable oxide layer, however after a few days, the bumps were reexamined and an oxide layer had formed. Further investigation into oxide layer formation must be done.

Post-machining Measurements

Different measurements were made to determine the quality and co-planarity of the diamond machined solder ball. A trace was done with a Tencor® surface profilometer. The trace shows a variation of $\pm 1 \mu\text{m}$ across the surface of a single planarized bump. *Figure 4* shows the surface roughness of a single solder bump for a μ BGA™ chip. The variation in height across one solder bump on a BGA ($\pm 1 \mu\text{m}$) was larger than across the μ BGA™ ($\pm .4 \mu\text{m}$).

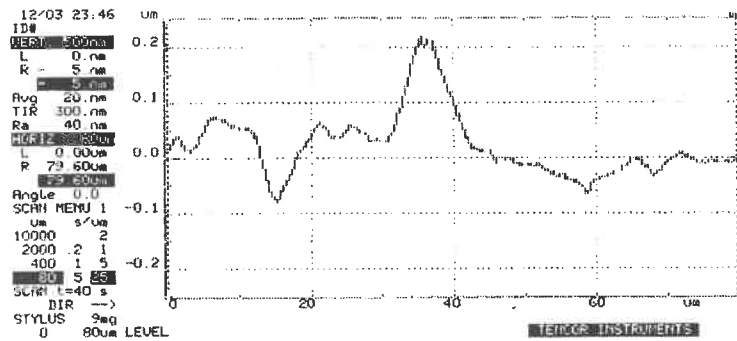


Figure 4: Surface roughness trace of μ BGA™ after diamond machining

In addition to doing surface profilometry, we measured each solder ball within the array on the CMM machine. Similar to the pre-machining measurements, the measuring probe was indexed along the rows of the array measuring each height. The results indicated that the co-planarity of the diamond turned surfaces was between 1-7 microns. Figure 5 shows a graph of the results.

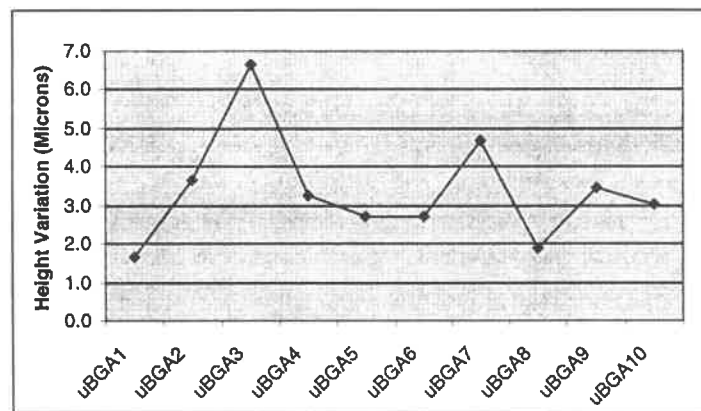


Figure 5: Maximum peak-to-valley height variations of the μ BGATM measured

In addition to the CMM measurements, the planarized μ BGATM were sent to a process metrology lab where they use optical profiling techniques to measure the topology of surfaces. The automated optical profiler expedites process control of large-format samples with accurate, non-contact 3-D data. The lab data shows the average co-planarity to range between 9-10 μ m. This data compares well with the co-planarity data from the CMM.

Conclusion

To date, this study has supported the hypothesis that planarizing BGA and μ BGATM chips can minimize the deflection of a contactor thereby improving the reliability of the contactor. The next step in the project is to investigate the effects of the oxide layer formation on the planarized surface. In parallel with this, designing rigid contactors using contact mechanics and Hertz elastoplastic solutions will be done.

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Investigation of the diamond machinability of newly developed hard coatings

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Introduction

Presently, electroless nickel is believed to be the hardest diamond turnable coating material, frequently used as a plating material for injection molds and high quality metal mirrors [1-4]. We have investigated the diamond machinability of new developed hard coatings deposited with a thickness of 10-20 μm by a PVD-process. The sputtered hard layers are superior to electroless nickel and even hardened steel. Taking this into account, a variety of new applications would be possible.

Experimental results

In a first approach we have investigated the diamond machinability of TiN-, TiAlN-, AlN- and CrN- coatings with varying nitrogen content. The coatings were deposited with a thickness of 20 μm on high speed steel plates and face turned with single-crystal diamond tools with a nose radius of 3 mm. The experiments were carried out on a Hembrug Super Microturn CNC lathe and on a Moore M 18 Aspheric Generator with a depth of cut and infeed of 5 μm . At constant spindle speed of 500 rpm the cutting speed varied from 12-45 m/min. The cutting- and thrust forces were recorded with a Kistler 9257 B piezo electric transducer. Tool wear was investigated with a scanning electron microscope (SEM) and by measuring the edge radius of the diamond with an atomic force microscope (AFM) [5]. Investigation of the surface topography and determination of roughness values were performed with a white light interferometer.

Fig. 1 shows the surface conditions of different diamond turned hard coatings after cutting distances of 130 m. Concerning hardness, the different coatings are comparable.

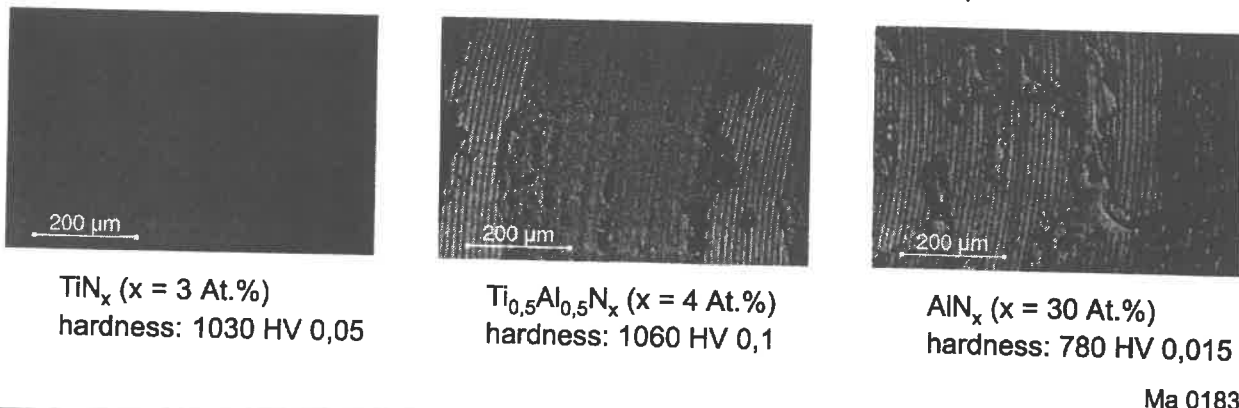


Fig. 1: Nomarski photographs of diamond turned hard coatings.

In case of TiAlN, AlN and CrN diamond turning leads to a partial disruption of the hard coatings exposing the feed marks on the steel substrate. SEM investigations of the used diamond tools revealed high abrasive tool wear.

Considerably higher surface qualities were obtained on TiN coatings. The influence of nitrogen content on the roughness R_a of the machined surface and the hardness of the coatings is shown in fig. 2.

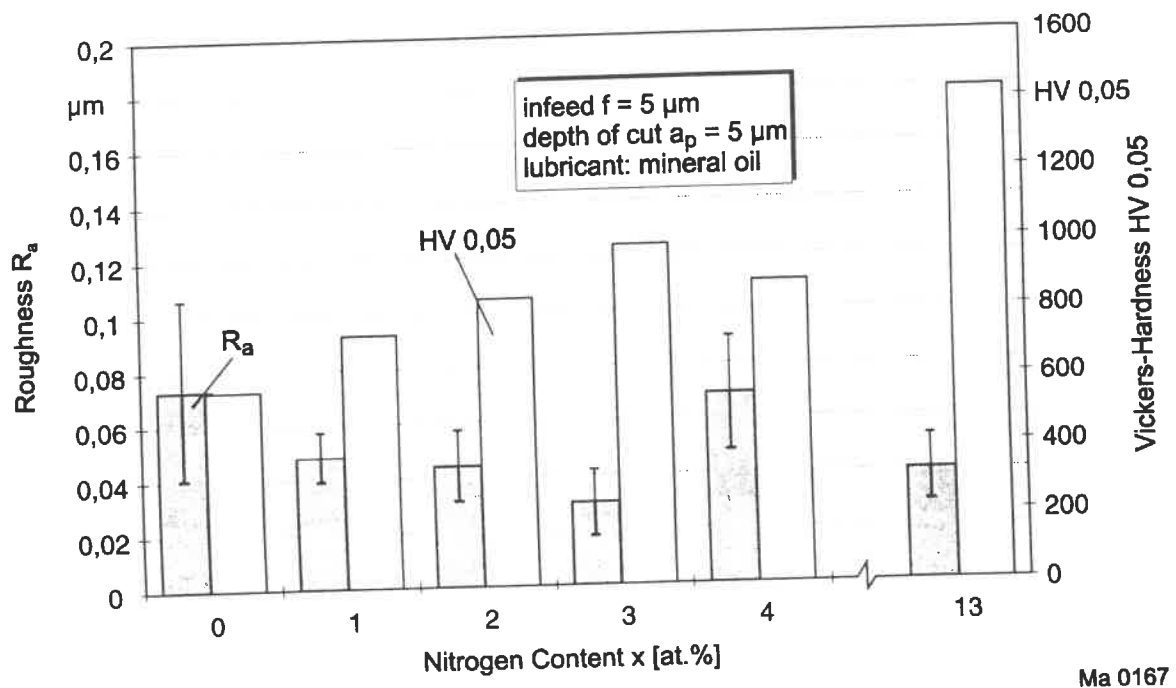


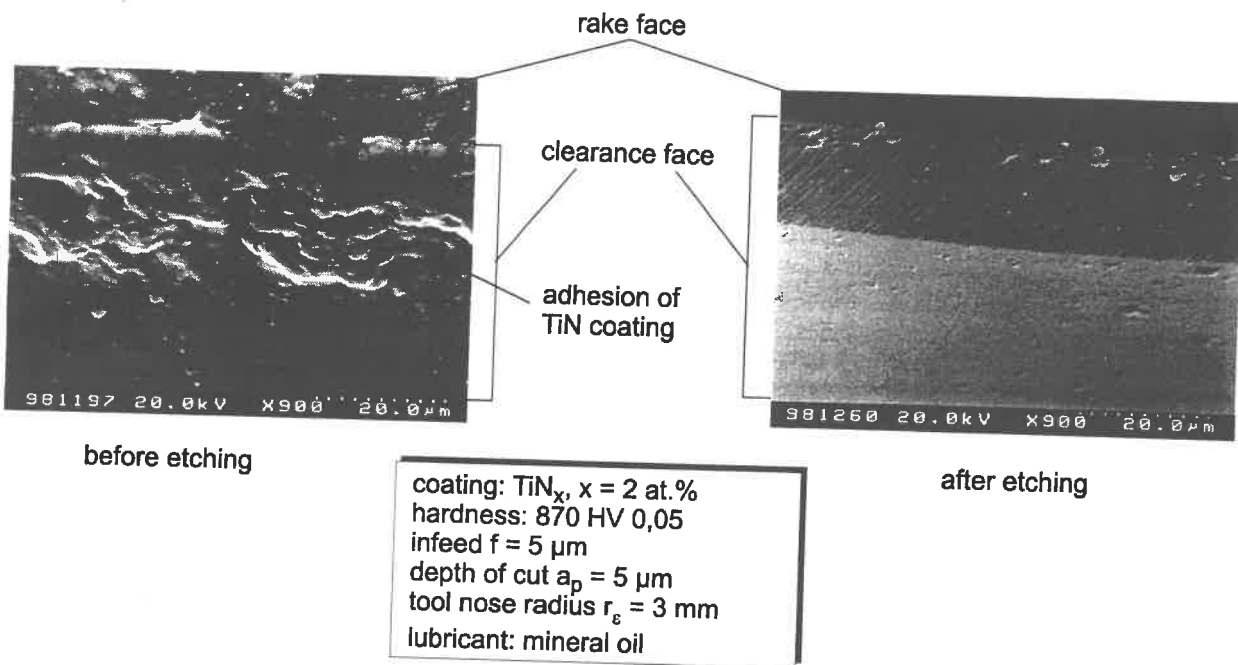
Fig. 2: Roughness R_a and hardness of the coatings for different nitrogen contents.

It may be noticed, that there is no correlation between the Vickers-hardness of the coating and the achieved roughness value R_a .

A roughness minimum was obtained for coatings with a nitrogen content of $x = 3$ at.%. In this case the hardness of the coating is even superior to hardened steel. In some experiments optical quality was reached with a roughness $R_a < 10$ nm. Moreover, the wear of the diamond tool is still tolerable after a cutting distance of 130 m. However, coating material removed during the turning process, adheres at the cutting edge which influences the material removal mechanisms. In order to avoid adhesion, further experiments with reduced cutting speed, different lubricants and with ultrasonic vibration of the cutting tool will be performed.

Tool wear was investigated by electron- and atomic force microscopy after cleaning the tools in sulfuric acid until any workpiece material adhering to the cutting edge was removed. Images

of a cutting tool before and after cleaning are presented in fig. 3. Although fine scratches on the wear land can be recognized, a sharp cutting edge is maintained.

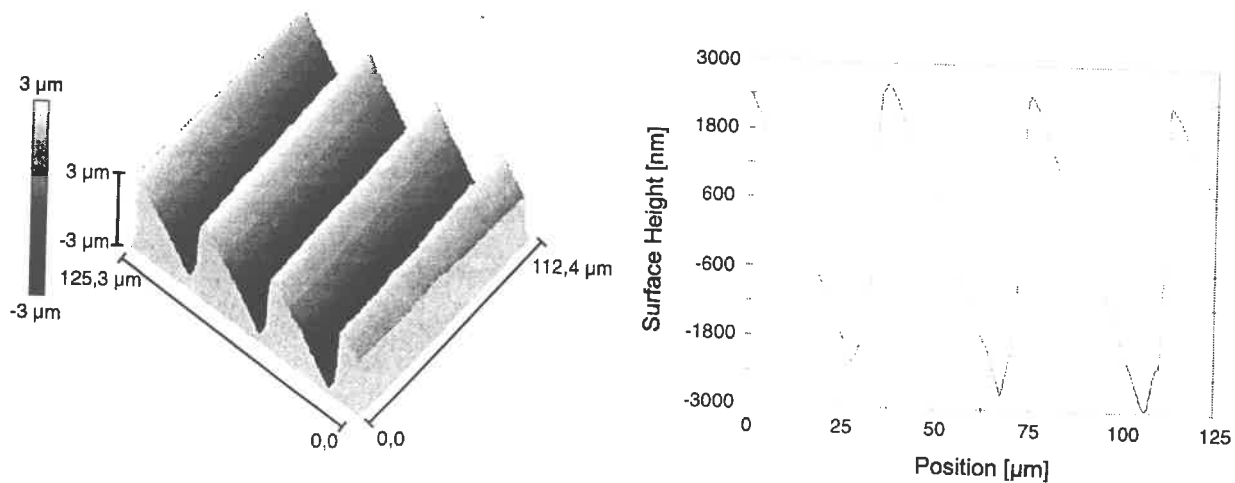


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Fig. 3: SEM-images of a cutting edge before and after etching in sulfuric acid.

Generation of microstructures in hard coatings

In order to verify the suitability of the new developed coating for generating microstructures, Fresnel structures were turned into a TiN coating with a nitrogen content of 3 at. % on a Moore M 18 lathe with a $5 \mu\text{m}$ radius tool. The diamond turned structure was investigated with a white light interferometer revealing a well defined groove geometry (cf. fig. 4).



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Fig. 4: Fresnel-structure generated in a hard coating.

Conclusion

We have investigated the diamond machinability of hard coatings deposited on steel substrates by a PVD process. While diamond turning leads to a partial disruption of TiAlN-, AlN- and CrN-coatings, TiN-coatings were not damaged and exhibit a high surface quality.

The best results were obtained with a nitrogen content of $x = 3$ at.% corresponding to a Vickers-hardness of 1000 HV 0.05. Over a cutting distance of 130 m, a surface roughness down to $R_a < 10$ nm could be achieved, while tool wear was still tolerable. We also succeeded in diamond turning a Fresnel structure into this coating using a diamond tool with a nose radius of 5 μm .

Acknowledgement

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Experiences in Micro-milling and Micro-drilling

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Introduction

The field of micromachining is expanding to meet the demand for increased functionality for smaller components – components that are frequently used in the aerospace, electronics and medical equipment. While no formal definition of micro-machined parts exists, it has been suggested that machined parts from millimeter to sub-millimeter in size and small parts that cannot be machined by conventional methods fall into this category [1,2,3].

The manufacturing methods vary widely to include methods with parallels to conventional machining such as milling, drilling, edm, grinding, punching and injection molding. Unconventional micromachining methods can include wire electrodischarge grinding, wire electrochemical grinding, micro ultrasonic machining, and electrolyte jet electrochemical machining [1]. Micro-machining should not be confused with micro-electromechanical systems (MEMS) which are microscopic in scale and typically fabricated by techniques used to manufacture integrated circuits (photolithography, etching, etc.). The manufacturing process for building MEMS machines has a very slow rate of growth (one silicon layer at a time), but micro-machining has high material removal rates and true 3-D capability.

The starting point for our work is micro-milling and micro-drilling. The objective is to apply our knowledge of conventional and high speed milling and drilling, in order to gain experience in and to understand the limitations, capabilities and processes of end mill tools and drills at a very small scale.

Test Bed

Micro-milling and micro-drilling have requirements beyond the capabilities of conventional machine tools. They include:

- the machine tool itself - very high spindle speeds are required to match the material removal rates of the small tools
- part fixturing - new challenges are represented as the parts are very small and difficult to fixture
- part size and part features - limited by the diameter of the tools, commercial twist drills are typically no smaller than 50 microns, spade drills are used for smaller holes,
- and due to the small sized parts, higher machine accuracy and resolution is required, typically 0.25 micron [1].

Our experiments were performed on a Makino¹ A55 Delta horizontal machining center. This machine has a 20,000 rpm ballbearing spindle and one micron linear scales. A 60,000 rpm Westwind airbearing spindle was adapted to the machine's spindle housing, as shown in Figure 1, for comparing the cutting performance of the two types of spindles and material removal properties at the very high speeds. By changing the work offsets, the same part program and setup can be used for milling or drilling with either spindle.

Experiments

The first set of micro-machining experiments were designed to test the following endmills:

1. four flute 1.0 mm endmill
2. four flute 1.6 mm ball endmill
3. single point diamond endmill.

In addition, a 50 micron commercial twist drill was tested. For each cutter, the chip load was established and a series of test cuts at various depths of cuts, for each spindle were performed. The initial test parts were copper, chosen out of convenience (had test parts on hand) and because of the interest in comparing the surface finish obtainable with a diamond end mill on the Makino machining center to that of a diamond turning machine. In subsequent experiments, aluminum and ferrous materials will be micro-milled and micro-drilled.

In addition, several features were machined: a 1.0 mm by 2.5 mm square and round posts and also a 4 mm square by 2.5 mm pocket. The features were machined with the four flute 1.0 mm endmill on two different samples: one at 20,000 rpm with the Makino spindle and one at 60,000 rpm with the Westwind spindle. The interest was in seeing if standard programming methods were sufficient to machine the small features or would trial and error be required to optimize the programming methods.

Experimental results

The simple experiments have provided an insight to the difficulties of micro level machining. One can see from Figure 2, the thin walled component whose wall thickness is 0.13 mm and 2.5 mm tall, a 20:1 aspect ratio. A resultant from any milling process is burrs, but on a small thin walled section, deburring is difficult, if not impossible on even smaller part features. While not visible at this magnification level, an even smaller burr exists along the top of the thin wall. From Figure 5, the height of the burr is equal to the depth of cut. The burrs from the previous passes were automatically cut off. This figure shows a corner from the 1.0 mm square post.

¹ Commercial equipment or materials are identified in order to adequately specify certain experimental procedures. In no case does such identification imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that equipment or material is necessarily the best available for the purpose.

Surface finish (R_a) from the test cuts were approximately 30% better with the Makino spindle at 20,000 rpm than the Westwind spindle at 60,000 rpm, and 60% better than the Westwind at 20,000 rpm. The conditions for the test cuts were: 1) the same endmills were used in both spindles, and 2) a constant chip load was maintained i.e. feedrate three times faster at 60,000 rpm. It was observed that the Westwind spindle slowed within the cut at 20,000 rpm, but not at 60,000 rpm. The torque is suspected to be greater at 60,000 rpm.

In addition, the part programming went smoothly: many light depths of cuts were needed to step down the walls, posts and pocket. Also, the parts were programmed to remove a complete layer of material, including finish pass, at every step down: a technique used in high speed machining of thin walls with an aspect ratio easily being 100:1 or more.

The drilling experiments have not been too successful, starting with the micro-drill quality. Shown in Figure 3, is a 50 micron commercial micro drill with 410 micron flute length. Upon inspecting five randomly chosen drills under a SEM, only one was free of the imperfections that we see in Figure 4 (note, the drill is the same in Figure 3 and 4).

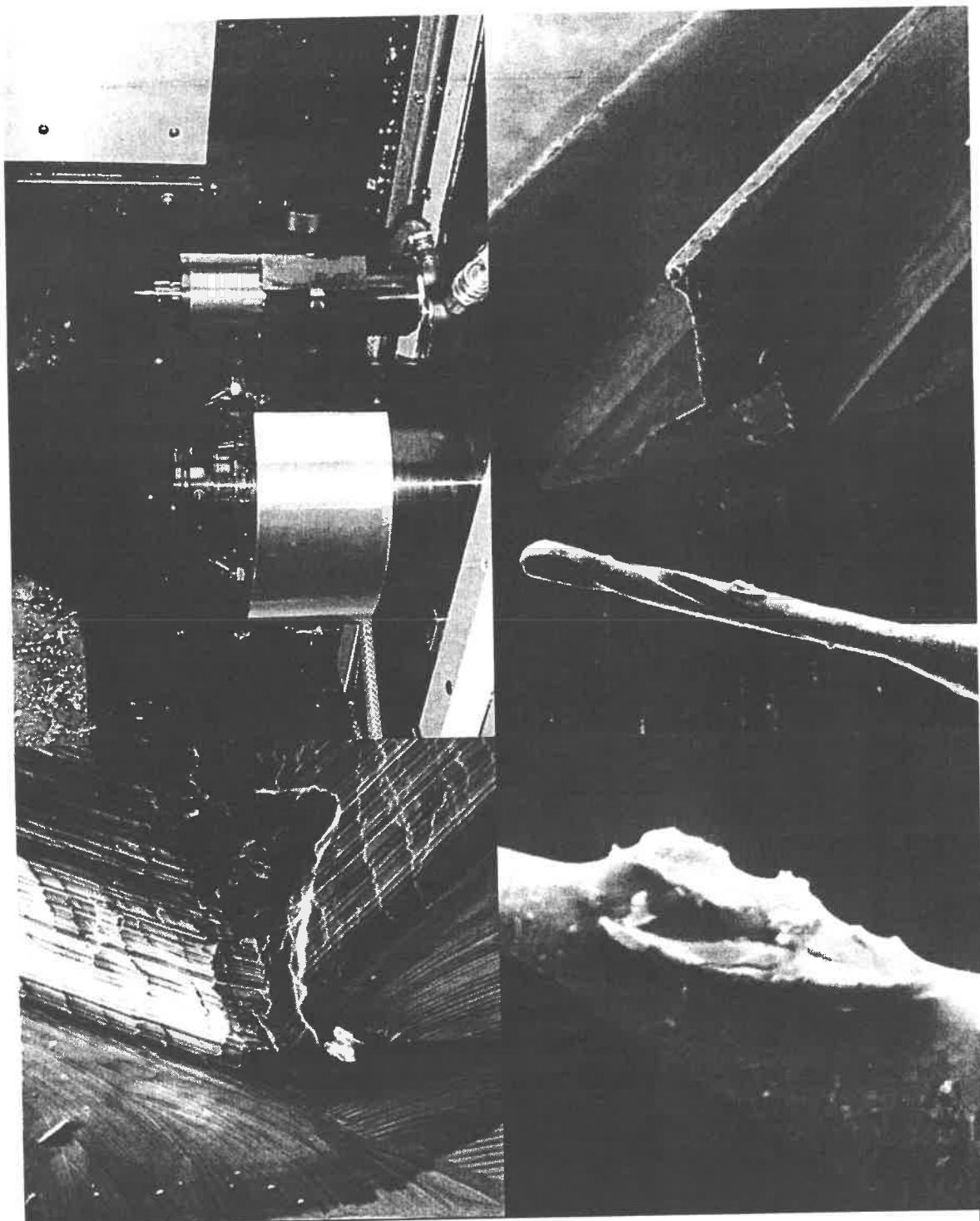
A micro indentation tool was manufactured for creating a small indentation on the part surface just as you would use a center drill to create a starting point for a conventional drill. This tool was used to create a series of ten indentations prior to our drilling. The first two holes were drilled to a depth of 300 microns without a problem, but the drill broke on the third hole. On inspection, we noted that the micro-indentation tool had a flat on its tip equal to twice the drill diameter: 100 microns. The reality was, the drill was wondering in the absences of a starting point thus leading to it's demise. Its been suggested that sensitive torque feedback control is necessary to minimize micro-drill breakage, but is difficult to instrument [1]. An experienced operator's touch and feel works well on manual machines.

Conclusions

The experience gained from these experiments will be used to shape new experiments: using smaller and different types of tools to machine smaller more intrigue features in aluminum and ferrous materials. Further work could also focus on the work piece fixturing issues and metrology of small parts.

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Figures clockwise from top left

- Figure 1, Dual spindle mounting, 20,000 rpm with 60,000 rpm mounted on top.
 Figure 2, Thin walled section 0.13 mm by 2.54 mm tall (25x magnification).
 Figure 3, SEM image of 50 micron commercial drill (156x magnification).
 Figure 4, Close up view of drill body imperfection (1560x magnification).
 Figure 5, Burr on corner of square post due to 60,000 rpm milling operation (230x magnification).

MECHANICAL MICROPROFILING MECHANISMS OF ADVANCED CERAMICS

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Keywords -- Grinding, Microprofiling, Advanced Ceramics, Scribe Test, Brittle Fracture Mode Cutting, Ductile Mode Cutting

Abstract -- For brittle materials like advanced ceramics, the most likely machining mechanism to occur is brittle fracture mode cutting. This results in substantial subsurface damages as well as considerable edge chipping. However, if a submicron depth of cut is chosen and a sufficiently high specific energy is applied, a ductile mode cutting, avoiding the issues mentioned above, may be achieved. To investigate the conditions resulting in ductile mode cutting, a combination of scribe and dicing experiments were performed on manganese zinc (MnZn) ferrite, a material used extensively for fabrication of read-write heads for magnetic storage. The material was chosen due to its availability in both polycrystal and single crystal structure and for its very low fracture toughness.

I. INTRODUCTION

Advanced ceramics are highly susceptible to crystalline microfracture. The root cause for this brittle behavior is the ceramic's crystalline structure which is formed by a combination of covalent and ionic atomic bonds. Therefore, when subjected to machining, brittle mode cutting is the most likely cutting mechanism to occur [1]. In brittle mode cutting, the contact between the cutting tool and the workpiece causes the initiation and propagation of microcracks. It results in both the formation of chips as well as subsurface damages due to microfracturing of the ceramic material. When mechanically machining microparts, this cutting mode is undesirable. For achieving the required surface and edge quality as well as for minimizing subsurface damages, ductile mode machining is required. This mode is based on a plastic deformation due to displacement activation in the crystal lattice. In any grinding process, a statistical number of grinding grit is engaged. However, the ductile cutting mechanism and its transition to brittle mode cutting is best studied when looking at a single abrasive grain. Scribe tests using a geometrically defined diamond stylus are most suitable for such an investigation. These stylus results are transferable to dicing processes and allow an optimization of the process parameters [2].

II. THEORETICAL CONSIDERATIONS

For ceramics, a theoretical approach for the relationship between the average depth of cut of a single grain h_{cu} and the process conditions has been established by Roth and Toenshoff [3]:

$$h_{cu} = \sqrt[3]{\frac{0.4 \cdot d_g \cdot \bar{x}^2}{\tan\left(\frac{\theta}{2}\right)} \cdot \frac{v_f}{v_c} \cdot \sqrt{\frac{a_g}{D}}} \quad (1)$$

The average depth of cut depends on the tool topography (diamond grain size d_g , cutting edge angle θ , average abrasive grain distance $\bar{x}$), the process parameters (cut speed v_c , feed rate v_f , blade depth of cut a_g), and the blade diameter D . An empirical factor (0.4) represents the degree of diamond grain exposure due to blade dressing.

The critical depth of cut $h_{cu,crit}$ for the transition between brittle fracture mode and ductile mode cutting was established by Bifano [4]:

$$h_{cu,crit} = \psi \cdot \left(\frac{E}{HV}\right) \cdot \left(\frac{K_{Ic}}{HV}\right)^2 \quad (2)$$

E is the material's Young's Modulus, K_{Ic} its Fracture Toughness, HV its hardness in Vickers, and ψ a surface damage factor assuming the degree of ductile mode cutting, typically set to $\psi = 0.15$.

These two equations provide a theoretical basis for choosing machining parameters which should yield ductile mode cutting conditions. Of particular interest is Bifano's equation which allows a prediction which average depth of cut at the grain level may not be exceeded to allow for predominantly ductile mode machining.

III. EXPERIMENTS

A. Probe Material Considerations

An advanced ceramic lending itself exceptionally well for cutting experiments is manganese zinc (MnZn) ferrite, a material which has been used extensively for the fabrication of read-write heads in magnetic storage. This material, possessing a very low fracture toughness, is available both in a polycrystal and single crystal structure. The characteristic material parameters are: Fracture Toughness $K_{Ic} = 0.8 \text{ MPa } \sqrt{\text{m}}$, Young's Modulus $E = 170,000 \text{ N/mm}^2$, and a Vickers Hardness $HV = 650$. Entering above mentioned parameters in equation (2) results in a $h_{cu, \text{crit}} = 63 \text{ nm}$.

Polycrystalline materials lend themselves particularly well for cutting experiments since the material's grain structure has a substantial influence on the location of the cut. A cut may follow grain boundaries („intercrystalline cut”), or may be a combination of both („intercrystalline cut”), may cut through the grain („intracrystalline cut”), or may be a combination of both („intracrystalline cut”), see Fig. 1. We expect intercrystalline material separation to be caused by a brittle mode cutting which involves crack propagation along the grain boundaries and results in grain pullout (Fig. 1a.). On the other hand, a transcrystalline cut may be caused either by brittle fracture mode cutting involving crack propagation through the grain (Fig. 1c.) or ductile mode cutting (Fig. 1d.). Ductile mode cutting requires a submicron depth of cut and a specific energy high enough to cause plastic material deformation [1], [4].

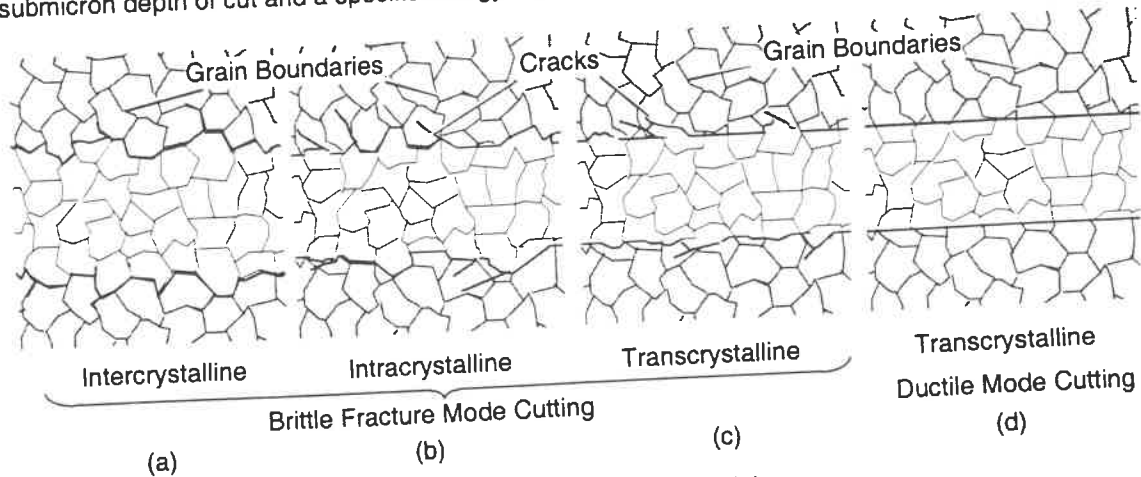


Fig. 1: Cutting mechanism classification for polycrystalline materials

B. Scribe Test

To determine the critical depth of cut for the transition between ductile and brittle fracture mode machining it is desirable to investigate the processes taking place between a single diamond grain of the dicing wheel and the probe during cutting. Since it is undetermined which diamond grain will actually engage in a cutting process, an experiment utilizing an actual dicing wheel does not look promising for performing this task. However, the cutting conditions at a single diamond grain may be simulated using a scribe test. In a scribe test, a diamond stylus is creating a scratch in a smooth and flat probe surface (Fig. 2).

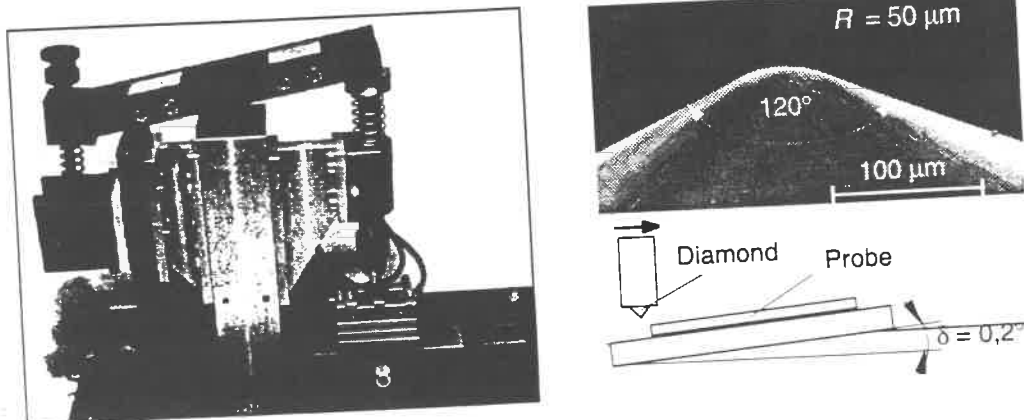


Fig. 2 (a): Scribe test stand

Fig. 2(b): Single scribe diamond, test principle

Due to a minute angle between the probe surface and the feed direction, the stylus penetrates the probe surface the deeper, the further it is fed. Such a test is expected to provide information regarding the phenomena when the critical depth of cut required for ductile mode cutting is exceeded. The

experiments were carried out using a test stand equipped with a 3-Component Force Transducer, (Fig. 2a.). The diamond stylus used in the experiments was cone shaped; the cone angle was 120° and the tip radius was $R_R = 50 \mu\text{m}$, (Fig. 2b.).

C. Dicing Experiments

A scribe test is expected to shed some light on phenomena occurring in the transition between ductile and brittle mode cutting. However, to verify that the selected cutting conditions do result in a ductile mode cut yielding optimal surface finish and edge quality, actual dicing tests are required. They are also needed to pinpoint the optimal depth of cut for ductile mode machining.

The dicing experiments were carried out using a high precision dicing saw equipped with an air bearing spindle. Room ambient, spindle coolant, and cutting fluid were held at a controlled temperature of $20^\circ\text{C} \pm 1^\circ\text{C}$. The dicing saw's maximum rotational speed was $40,000 \text{ min}^{-1}$, the feed speed range was $0.01 - 75 \text{ mm/s}$. The dicing wheels used in the experiments were resin bonded hubless type blades with average diamond grain sizes of 5, 15, or $30 \mu\text{m}$, mounted between a pair of flanges. Table I summarizes the test conditions for the experiments performed.

TABLE I
Dicing Experiments Process Parameters

Probe material	MnZn ferrite, polycrystalline	MnZn ferrite, monocrystalline
Blade type / binder	Hubless; resin	Hubless; resin
Diamond grain size	5, 15, and $30 \mu\text{m}$	5, 15, and $30 \mu\text{m}$
Feed rate	$0.1 - 10 \text{ mm/s}$	$0.1 - 10 \text{ mm/s}$
Cut speed	86 m/s	86 m/s
Cut depth	0.5 mm	0.5 mm
Feed direction	Down cut	Down cut
Coolant	Water	Water

III. EXPERIMENTAL RESULTS

A. Scribe Test Results

Fig. 3 depicts a typical scribe test result performed with polycrystalline MnZn ferrite as the probe material. It allows to determine the cutting mode as a function of the stylus's nominal depth of cut. The scribe test differs in test parameters from dicing in linear feed rate ($v_s = 0.1 \text{ mm/s}$, which is only a fraction of the dicing cut speed), the lack of coolant, and a substantially different diamond geometry. Due to these differences, the depth of cut is substantially different. Nevertheless, a transition from ductile to brittle fracture mode cutting and machining phenomena associated with it may clearly be observed.

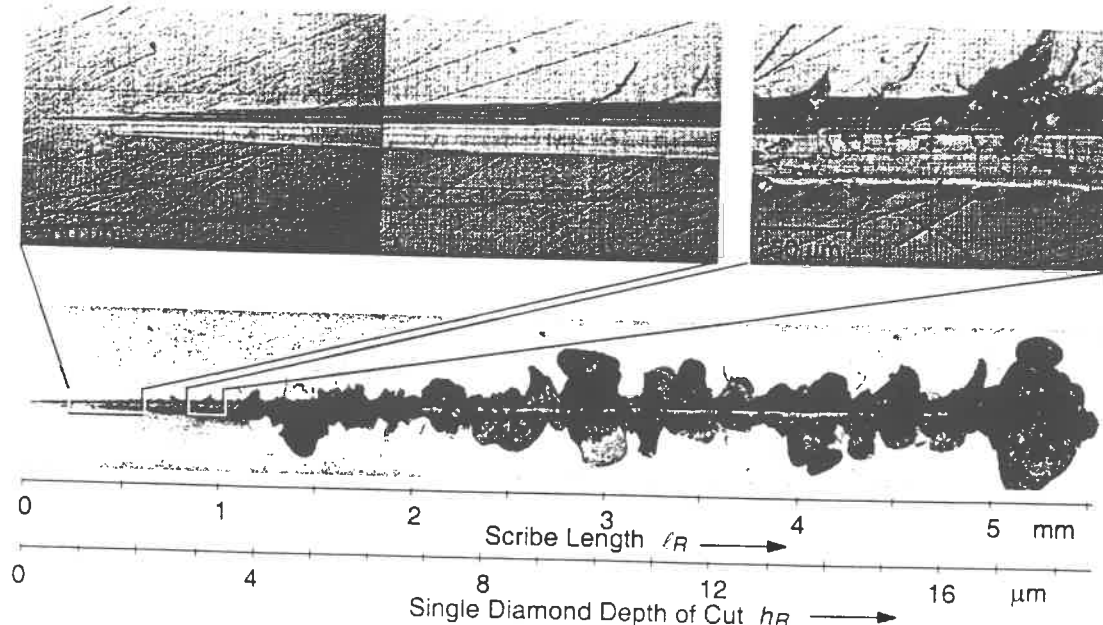


Fig. 3: Scribe mark in the surface of a polycrystalline MnZn ferrite probe

After engaging, the stylus created a crack-free scribe mark, first through ductile microploughing and microcutting and then by means of ductile material removal, creating a burr at the edge of the scribe mark. When reaching a nominal depth of cut of $1.5\text{ }\mu\text{m}$, first damages through brittle fracture occurred. The resulting cracks were crescent shaped. They originated either in the plastically deformed edge area or at the bottom of the groove. These sheer cracks started out nearly tangential to the stylus feed direction and then curved away. With a further increase of the scribe mark depth, the formation of lateral cracks within the scribe mark groove occurred. Ultimately, a system of crisscrossing and intersecting cracks was created, resulting primarily in grain pullouts within the groove as well as chips at the edge. At a nominal depth of cut of approximately $3.5\text{ }\mu\text{m}$, any correspondence between the scribe mark's nominal depth of cut and the actual scribe geometry was lost. The material shattered, resulting in a groove much larger than the equivalent stylus geometry and was characterized by substantial grain pullout and edge chipping.

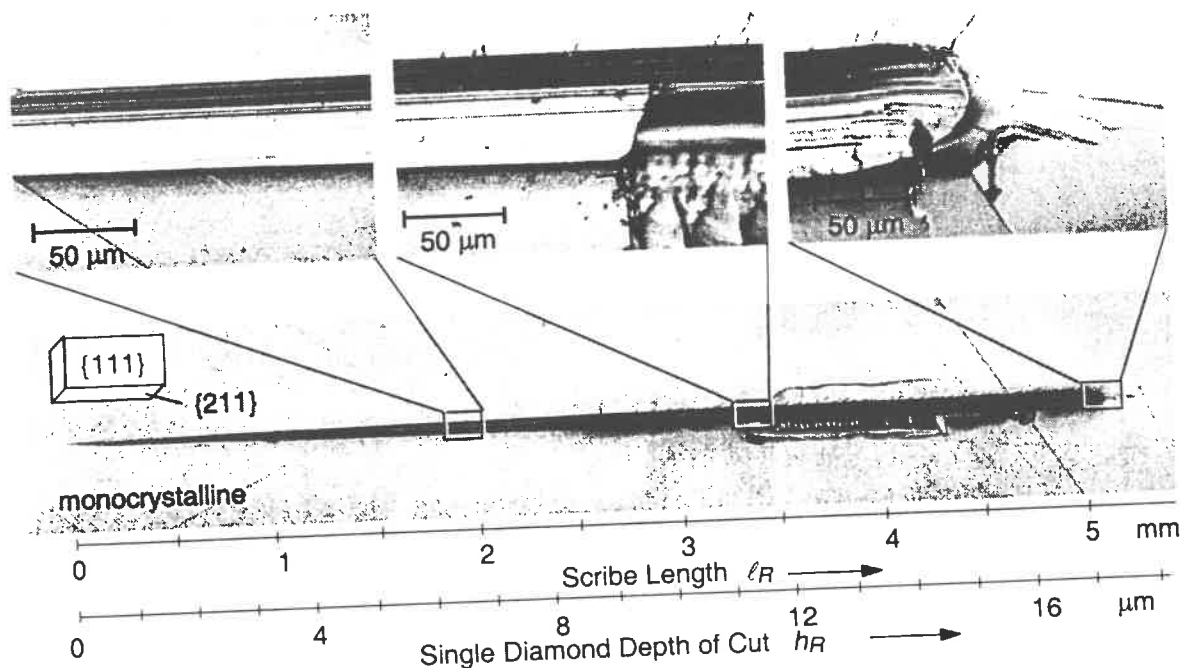


Fig. 4: Scribe mark in the surface of a single crystalline MnZn ferrite probe

Fig. 4 shows the scribe mark in a monocrystalline MnZn ferrite. The feed direction has been chosen in $\langle 110 \rangle$ direction, a dominant crystal direction represented by the intersection of the $\{111\}$ plane (probe surface) and the $\{211\}$ plane. Under these conditions, a ductile cut scribe mark can be generated up to a nominal depth of cut of app. $9.5\text{ }\mu\text{m}$. At this point, edge cracks typically running at an angle of 60° away from the scribe mark edge start to occur. Catastrophic material failure, caused by brittle fracture, first appears at a nominal depth of cut of app. $11\text{ }\mu\text{m}$. The surface crack follows the $\{211\}$ plane, while the chipping shows characteristically a material breakout of a glide plane due to the direction of the lateral cracks. The propagation of these cracks below the surface corresponds to the $\langle 211 \rangle$ direction and the $\{111\}$ crystal glide planes. These are characterized by a relatively large atomic distance with only a few manganese or zinc ions on interstitial places. The end of the scribe mark demonstrates the plastic deformation capability of the monocrystalline material, despite its brittleness and low fracture toughness. As a result of the compression stress in front of the scribe diamond, the material is displaced by microploughing and remains like a bow wave frozen in time.

For both the polycrystalline and monocrystalline MnZn ferrite, Fig. 5 depicts the scribe normal force F_n , the tangential force F_t , as well as the force ratio as a function of the depth of cut. Due to the great negative cutting angle, a high normal force and a great pressure below the rounded diamond stylus result, as long as the depth of it is small. As a consequence, a substantial portion of the groove formation apparently was done through microploughing. This is most obvious in case of the polycrystalline MnZn ferrite, reflected by the force ratio $q = F_n/F_t$, starting out high and decreasing with a growing nominal depth of cut.

The decrease of the force ratio indicates a growth of the tensile stress compared to compressive stress with an increasing depth of cut. This effect is caused by a change in the negative cutting angle α , see Fig. 6. At the beginning of the cut, for a diamond stylus depth of cut $h_R = 0$, α is -90° (Fig. 6a.). At a depth of cut of $4\text{ }\mu\text{m}$, due to the radius of the scribe diamond, α has increased to -66° . This increase, plus the increase of the cut cross section, results in growing forces causing an expansion of the crack length.

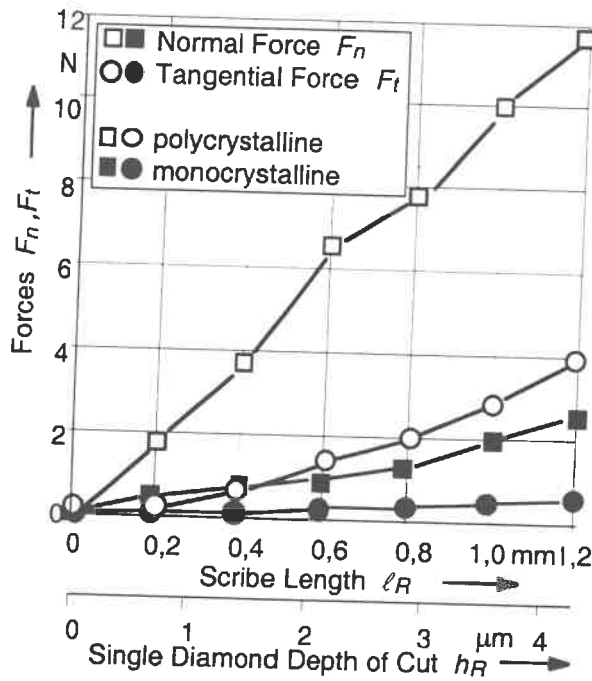


Fig. 5 (a): Scribe forces (spherical tip diamond)

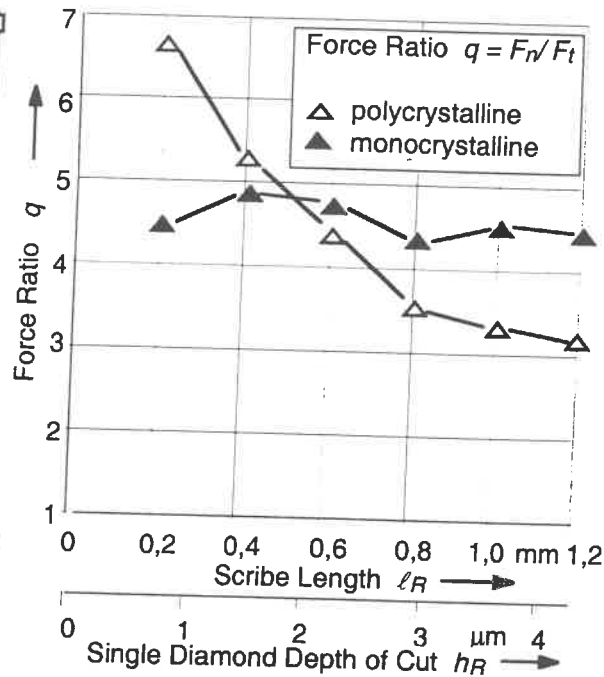


Fig. 5 (b): Scribe forces (spherical tip diamond)

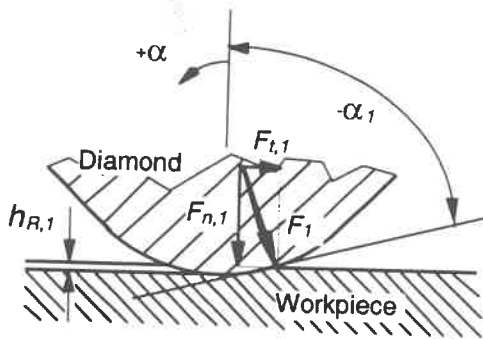


Fig. 6 (a): Negative cut angle α , low penetration

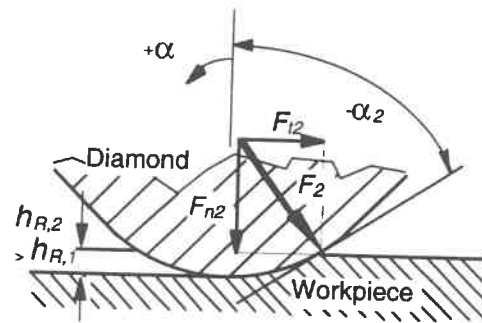


Fig. 6 (b): Negative cut angle α , high penetration

Using monocrystalline instead of polycrystalline MnZn ferrite, both the normal and tangential forces are on a considerably lower level. The crystal displacements are two-dimensional and undisturbed by grain boundaries, with cracks propagating along the preferential crystal slip planes. In case of the polycrystalline material, the force ratio is decreasing; it is nearly constant ($q = 4.5$) for monocrystalline material due to a nearly exclusive ductile mode groove formation over the complete scribe mark length.

The presence of microploughing in scribe testing indicates that there are limitations in the analogy between diamond stylus scribe testing and actual grinding. Microploughing is a phenomenon typically avoided in precision grinding. Nevertheless, the scribe test demonstrates impressively phenomena taking place in the transition phase between ductile and brittle mode cutting.

B. Dicing Results

The best dicing results were achieved with a diamond grain size of $5 \mu m$ and a feed rate less than $0.1 mm/s$. Fig. 7a. depicts the polycrystalline material machining result. The edge is chipfree. To substantiate the cutting mode, chips were filtered from the coolant and subjected to a SEM/EDX analysis. Fig. 7b. shows an SEM micrograph of helical chips, which clearly can be distinguished, and are an indication for the presence of a ductile cutting mode. The plastic chip deformation requires a higher level of specific energy than the crack propagation, due to the movement of the crystal displacements. This is given through the low feed rate and a large number of active diamond grains with a small diamond grain size. An EDX analysis (not shown) verified the presence of Fe, Mn, and Zn in the chips. Using equation (1), the theoretical value for the average single grain cut width may be calculated. For $v_f = 0.1 mm/s$, $v_c = 86.4 mm$ and $a_g = 0.5 mm$ the depth of cut value is $114 nm$. Fig. 7b. demonstrates a good match between theoretical and experimental data.

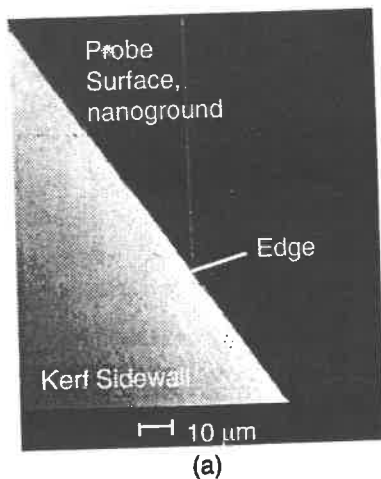


Fig 7 (a): SEM-image of a probe edge (MnZn ferrite, poly)

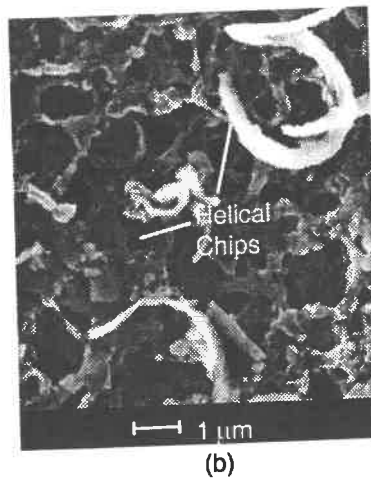


Fig 7 (b): SEM-image of the abrasive products

Fig. 8 and Fig. 9 depict atomic force microscopy (AFM) images of a polycrystalline and a monocrystalline MnZn ferrite sample where the edge is measured under an angle of 45°. Each figure shows a 3-D plot of the edge topography and the matching cross section. While the SEM-image reveals areas with a nearly ideal edge contour, the higher resolution of the AFM demonstrates transcrystalline chipping in the submicron range for the polycrystalline material (Fig. 8). These defects do not appear in case of the monocrystalline material (Fig.9) .

As in case of the scribe test, although much smaller, the cutting process results in some burring. The compressive stress during machining leads to a camber at the edge, which is about 40 to 50 nm in height, see Fig. 9.

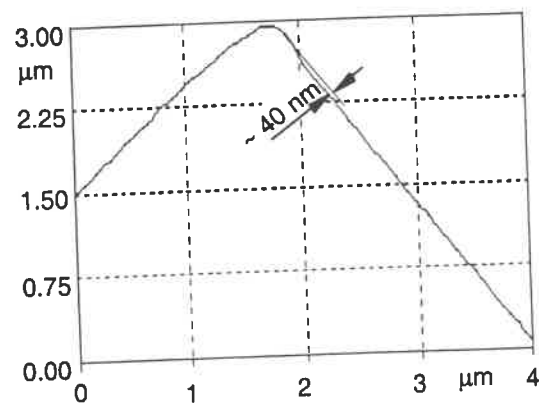
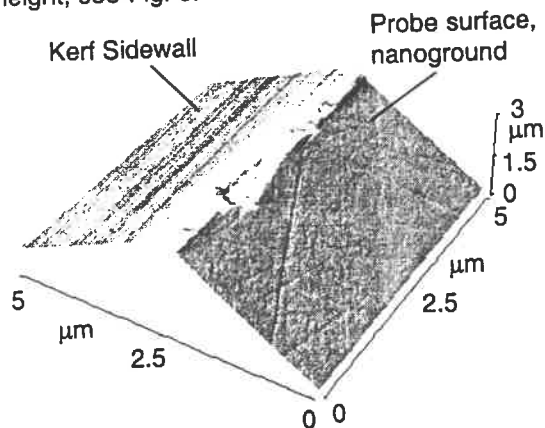


Fig. 8: AFM-image of an edge at polycrystalline MnZn ferrite

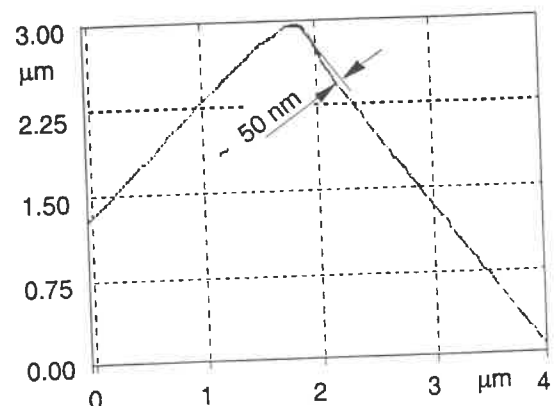
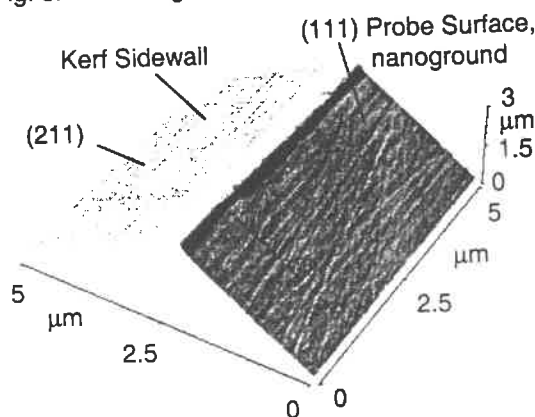


Fig 9: AFM-image of an edge at monocrystalline MnZn ferrite

IV. CONCLUSION

The requirement to generate a damage-free surface calls for ductile mode cutting. This is particularly important if chip-free edges are demanded. A scribe test was used to analyze the cutting mechanisms of a

single diamond grain and the phenomena occurring in the transition phase between ductile and brittle mode cutting. The results of the scribe test are qualitatively transferable on the dicing process, where the main conditions for ductile machining are a small depth of cut and a small diamond grain size. Achieving ductile mode cutting requires to choose an appropriate depth of cut, typically in the range of 100 nm or below. Theoretical predictions regarding this depth of cut were verified by experimental data, showing a good agreement between both. Since a plastic deformation of brittle materials requires a higher force than brittle fracture, a sufficiently high specific energy has to be provided.

The crystal lattice influence can be shown by the comparison of monocrystalline and polycrystalline material in scribe tests as well as in dicing experiments. As a result, the susceptibility of polycrystalline material to chipping is much greater than for monocrystalline material. Nevertheless, it is possible to create edge areas without any cracks and submicron chipping for polycrystalline material, but it is much easier to do so in monocrystalline material when properly choosing the crystal planes. In this case the glide planes are undisturbed by the grain boundaries.

During machining, the material is under a great compression stress. As a result, the cutting process forms a camber at the edge of the scribe mark kerf as well as at the edge of the diced kerf. For dicing, the camber height is in the range of approximately 50 nm.

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ULTRAPRECISION MACHINING OF METAL MATRIX COMPOSITES

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INTRODUCTION

Metal matrix composites (MMCs) were known for their synergistic properties; however, sensitive cost and fabrication challenges including machining were to be overcome for successful applications of these composites. Literature survey showed that finishing processes such as grinding and abrasive blasting were utilized to marginally improve the surface finish of machined MMC samples. The model for critical depth, d_c , associated with ductile regime in microscratching was proposed [Bifano et al, 1993]:

$$d_c = 0.15 \frac{EK_{cs}^2}{H^3} \quad (1)$$

where E is the Young's modulus, K_{cs} is the critical surface fracture toughness, and H is the hardness. For SiC, $E = 382\text{--}475$ GPa, $K_{cs} = 2.5\text{--}3.5$ MPa.m^{0.5}, and $H = 24.5\text{--}25.0$ GPa (Ratterman and Cassidy, 1991). Thus, the calculated range for d_c is 0.023–0.059 μm . This range for microscratching, however, does not agree with the critical grinding depth 0.6 μm for SiC [Bifano et al, 1988]. Grinding of Al/SiCp with diamond wheel at an infeed depth of 15 μm caused brittle fractures of the particles [Chandrasekarn and Johansson, 1997]. Ultraprecision machining of Al/SiCw with single crystalline diamond at 10 μm depth of cut also caused the whiskers to pull out, rotate, or "cut directly" [Yuan et al, 1993].

Despite the research efforts, fracture of the brittle SiC reinforcement due to machining has been a problem that requires further study. This paper investigates the material removal mechanism due to ultraprecision machining of cast aluminum alloy A359 reinforced with SiC particles. The objectives are to:

1. Implement the ductile regime machining technique to both the matrix metal and the reinforcing ceramic to produce high quality surface, and
2. Study the wear of both poly- and single-crystalline diamond tools

EXPERIMENT

This study used the A359/SiC/10p composite. The cast aluminum matrix contained Al 9.27wt% Si 0.15 Fe 0.55 Mg; and the 10 vol% SiC reinforcing particles had their median 9.3 ± 1.0 μm , 94% particles ≥ 3 μm , 3% particles $\geq 19\mu\text{m}$. Some samples were solution heat treated at 540°C for 14 hrs, water quenched, then peak aged at 155°C for 5 hrs. The respective hardness for the MMC in the as-cast condition (F) and aged condition (T6) was 71 and 92 RF. Both continuous and interrupted facing of short MMC coupons were performed on a commercial ultraprecision machine that had 2.5 nm positioning accuracy. Both single crystalline diamond tool (SCD, with (100)-rake face and 20–70 nm edge sharpness), and polycrystalline diamond tools (PCD, with 0.5 μm grains, 500–750 nm edge sharpness) were used for the facing operations. All feedrates were normalized as the percentages of tool nose radii. Some samples were machined at constant depths of cut of 0.2, 0.6, or 1 μm . Taper cut in other facing operations was programmed so that depth of cut varied continuously from 0–0.2 μm . Machined surfaces were analyzed with an atomic force microscope (AFM), a Form Talysurf profilometer (FT), or a laser interferometer. They were also observed directly, or

etched with Keller's etchant to clear the smearing aluminum on the surface, using a scanning-electron microscope (SEM).

RESULTS

A parallel study of ultraprecision machining of copper was completed. A mirror surface of <5 nm Ra was achieved using the same process parameters and tooling. The analysis showed that tool material significantly affected the surface finish, followed by the feedrate and tool radii. The effects of material conditions and cutting speeds, however, were less pronounced.

The calculated surface finish agreed with the measured surface finish for copper, but not the MMCs. Shattered SiC particles and tool wear degraded the surface finish of MMC samples. A PCD tool produced a smoother MMC surface than that from a SCD tool (Fig. 1). Examination using SEM and AFM showed that the surface of the samples were smeared with the soft aluminum matrix when machined with a PCD tool, while different phases and broken SiC particles were visible on MMC samples machined with a SCD tool. The surface roughness, therefore, varied greatly depending on whether the measurement was confined within an aluminum phase or the SiC-containing eutectic phase.

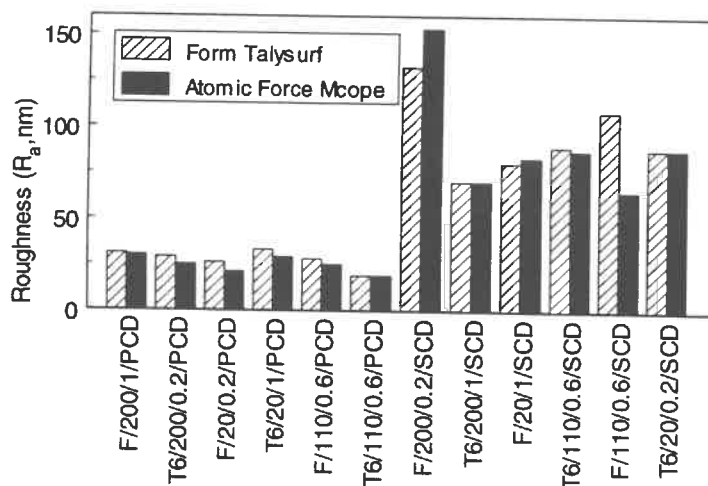


Figure 1. Comparison of surface roughness for ultraprecision machined MMCs. Facing A359/SiC/10p at feed = 0.5% nose radius ($\mu\text{m}/\text{rev}$). The sample's nomenclature indicates Temper /Cutting speed (m/min) /Depth of cut (μm) /Tool material.

Ductile regime machining of the hard SiC particles was confirmed by examining the as-machined surface, the cross sectioned subsurface, or the etched surfaces. At depths of cut below $0.2 \mu\text{m}$, some SiC particles were machined in the pseudo ductile mode (Fig.2), but some were shattered brittly. Shattering of a SiC particle by a diamond tool was due to:

- Deeper effective depth of cut as the result of particle rotation after engaged in cutting. The rotation was possible due to inadequate interfacial strength between a particle and the surrounding matrix, or plastic deformation of the matrix.
- Unfavorable orientation of that SiC particle with respect to the cutting tool.
- Defects in the SiC particles.

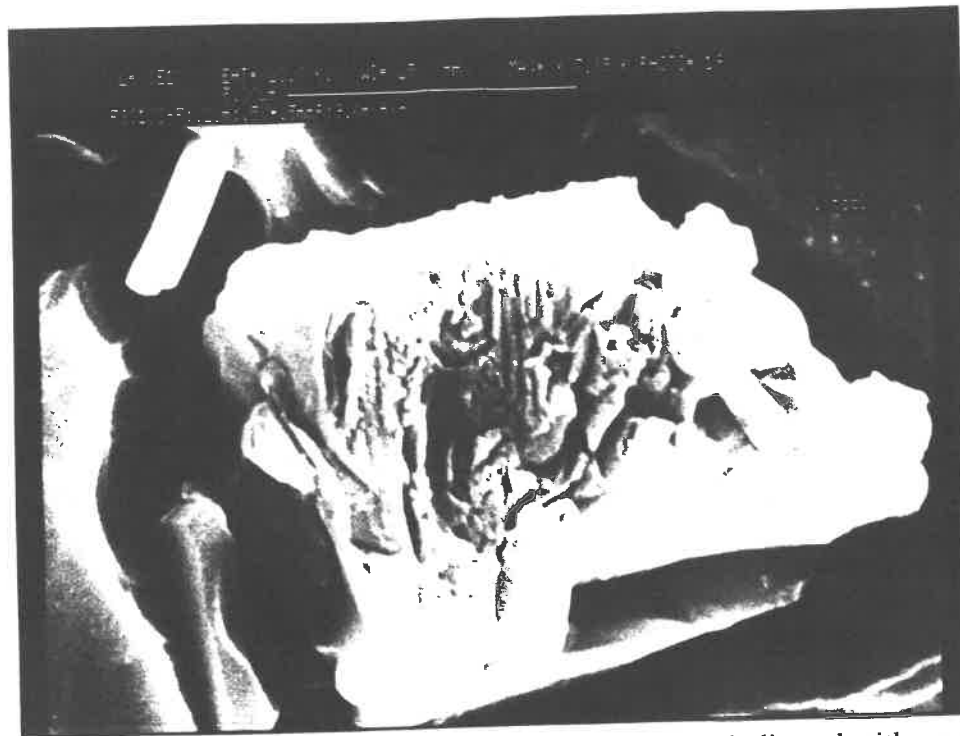


Figure 2. Pseudo-ductile mode of micro machining in SiC as indicated with vertical tool marks. Notice the micro-cleavages on the surface of the machined particle. Facing A359/SiC/10p, SCD tool with 0° rake, 9.6 m/min speed, 2.5 $\mu\text{m}/\text{rev}$ feedrate, 0.2 μm depth of cut. Etched with Keller's etchant.

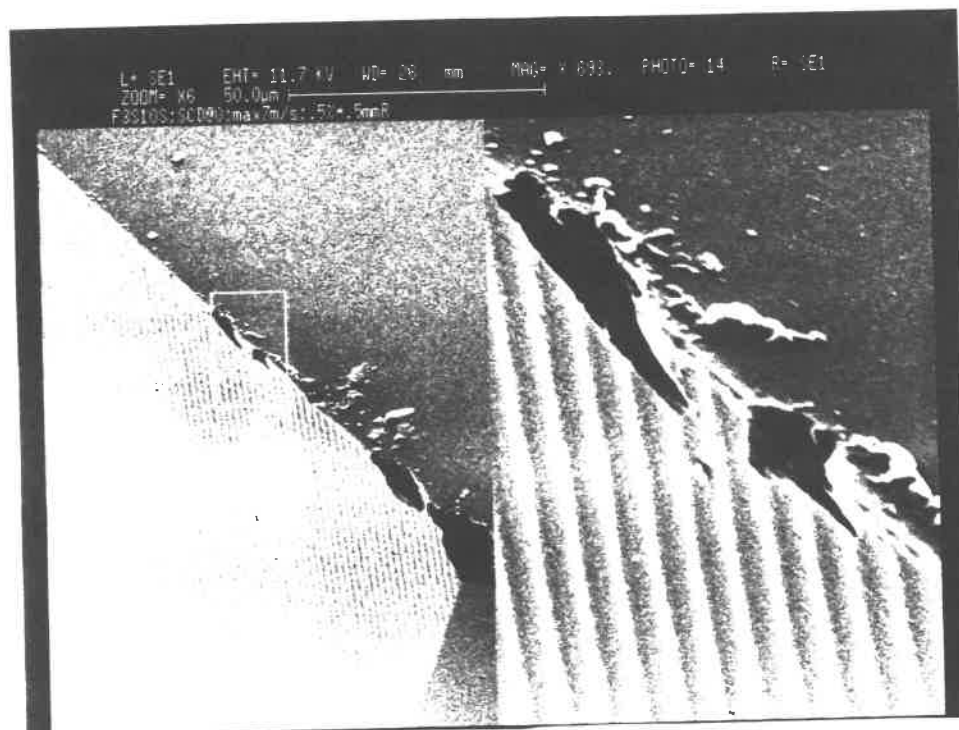


Figure 3. Substantial wear of single crystalline diamond tool. Notice the chipped off edges. Facing A359/SiC/10p, SCD tool with 0° rake, 420 m/min maximum speed, feed 2.5 $\mu\text{m}/\text{rev}$, 1 μm depth of cut.

The orientation of a brittle substrate contributed to different fracture mode [Shibata et al, 1995]. The SiC reinforcement had either diamond structure (β -SiC), hexagonal or rhombic structure (α -SiC). The critical resolved shear stress, on a crystalline plane of SiC due to the cutting action, was directly proportional to the Schmid factor $\cos\lambda\cos\phi$, where ϕ and λ were the orientations of the slip plane and slip direction. It was postulated that an ideal ductile mode happened when the cutting shear stress was parallel to both the slip plane and the slip direction, otherwise a pseudo ductile mode with micro cleavages was seen. Equation (1) did not have the orientation factors and was process independent. This probably was the reason for the variation of published critical depths for ductile regime machining of SiC.

Scanning electron microscopy examination showed "normal" abrasive wear of a PCD tool, but substantial abrasive wear of a SCD tool. Microchipping and surprising wear of a SCD tool is shown in Fig. 3, in which the appearance is similar to diamond tool wear when machining steel. A combined mechanism of diffusion and abrasive wear of SCD was possible when machining MMCs containing SiC particles.

CONCLUSIONS

Ultraprecision machining of A359/SiC/10p was studied. This investigation showed:

1. A PCD tool smeared the aluminum on the machined surface (20-30 nm Ra), but a SCD tool cut the surface effectively to reveal different phases (60-150 nm Ra).
2. Abrasive wear of PCD was seen, but a suspected diffusive-abrasive wear of SCD was observed.
3. Pseudo ductile mode machining of SiC particles was observed. Proper orientation of a particle and depth of cut below 0.2 μm were the sufficient conditions for ductile mode machining.
4. A revised model for ductile machining should include the crystallographic structures and orientation factor of a ceramic reinforcement.

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Vibration Cutting Method of OD-Blade With Atomized Working Fluid

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1. Introduction

Fixed diamond grains and the working fluid with high lubricant and efficient cooling performance in the machining region are used in OD-Blade cutting which is one of the precision processing methods for hard and brittle materials. This OD-Blade cutting method, however, has no enough eliminating effect of cutting chips so that the loading and attrition wear of the diamond grains are caused. Consequently, the conditions of the diamond grains during cutting have been related to a deterioration of cutting performance of OD-Blade. Therefore, in this method, it is an important factor to supply a large quantity of the working fluid. It is, however, closely connected with serious problems in not only an increase of waste fluid but also an environmental deterioration of the earth⁽¹⁾⁽²⁾.

This paper proposes a new cutting technique of OD-Blade which can improve the machining efficiency and accuracy, and make a remarkable decrease in quantity of the working fluid. In order to enable to accomplish these requirements, the authors have developed the new method of OD-Blade cutting which make an application the vertical vibration to the workpiece during cutting, in addition to supply the atomized working fluid spouted a minute quantity of the working fluid by means of the compressed air. This paper describes the results of a basic experiment and an investigation under a series of conditions.

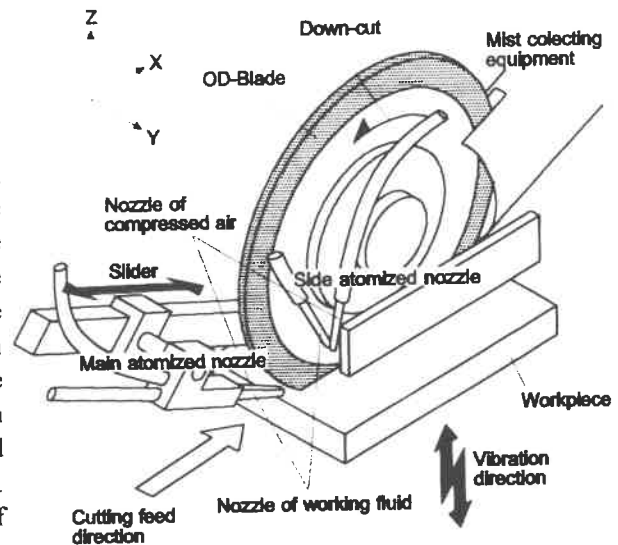


Fig.1 Model of vibration cutting using atomized working fluid

2. Experiment

2.1 Supply method of the atomized working fluid

A model of the vibration cutting method with the atomized working fluid to establish this study is shown in Fig.1. In this method, a minute quantity of the working fluid supplied from the fluid-nozzle is atomized by means of the compressed air from the air-nozzle, and this is supplied to the machining region with the compressed air. In addition, the atomized nozzle made up of the fluid- and air-nozzles set in the front (Main atomized nozzle) and both sides (Side atomized nozzle) of OD-Blade. The set angle of the main atomized nozzle especially equals the tangent angle in maximum central angle at contact arc between OD-Blade and the workpiece, and

Table 1 Main experimental conditions

OD-Blade, Size	D	$\phi 125 \times 1 \text{ mm}$
Type		Metal-bond, SD#200
Blade peripheral speed	V_s	1963 m/min
Slicing mode		Down-cut
Depth of cut	c	11.0 mm
Feed speed of workpiece	$\dot{X}$	40 mm/min
Frequency	f	20 Hz
Amplitude	a	15 μm
Workpiece, Size		Soda glass, $50 \times 60 \times 10 \text{ mm}$
Grinding fluid		Water soluble coolant (50:1)
Conventional nozzle	Q	3811 g/min
Main atomized nozzle		
Flow rate of working fluid	Q_m	10 ~ 40 g/min
Pressure of compressed air	P_m	0.05 ~ 0.20 MPa
Side atomized nozzle		
Flow rate of working fluid	Q_s	5, 10 g/min
Pressure of compressed air	P_s	0.10 MPa

can move in front or in back from the machining region as keeping this angle.

2.2 Experimental method and conditions

In this experiment is used a slicer equipped with NC-controller³⁾ and set up with the atomized nozzles, and the piezo-translator is used as a vibrator in the vibration cutting. The cutting characteristics is evaluated through the loading change in the driving current of the spindle motor, the wafer roughness measured by a surface roughness tester and the chipping amount in the surface of cutting groove. The main experimental conditions are as shown in Table 1.

3. Cutting characteristics with atomized working fluid

3.1 Examination for set point and pressure of compressed-air in main atomized nozzle

Fig.2 shows the behavior of the loading current in case of changing the set point of the main atomized nozzle by a slider. And, the space between a dead point of the main atomized nozzle and the machining region at the zero point in Fig.2 is 4.5mm. According to this result, the loading current increases as the space increases without regard to the cutting method.

Furthermore, Fig.3 shows the behavior of the loading current in case of changing the pressure of the compressed air from 0.05 to 0.20MPa. From this result, it is clear that the loading current in not only the non-vibration cutting but also the vibration cutting decreases as the pressure increases.

The behavior of the loading current for the changing of the set point and the compressed pressure at the main atomized nozzle is able to examined by attention paid to the behavior of the flow velocity of the atomized working fluid. When supplying the atomized working fluid, it is thought

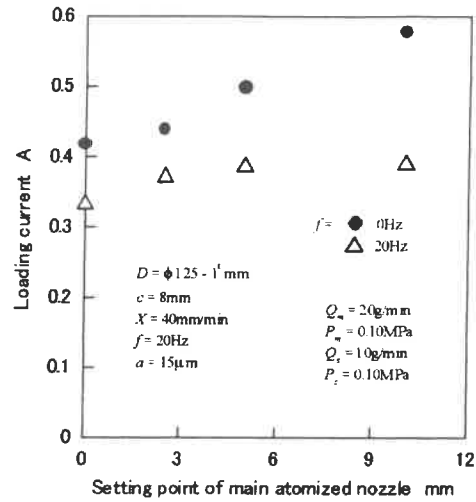


Fig.2 Behavior of loading current when changing in setting point of main atomized nozzle

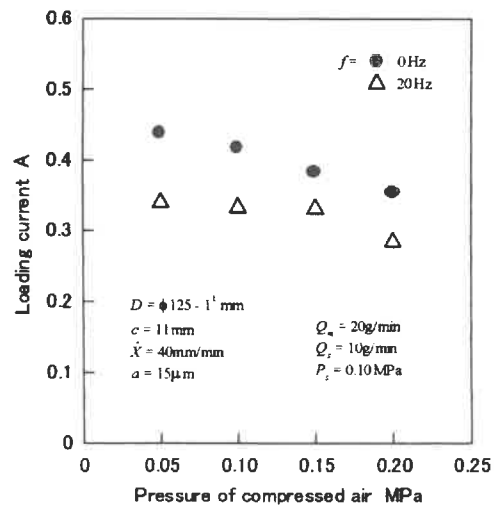


Fig.3 Behavior of loading current when changing in pressure of main compressed air

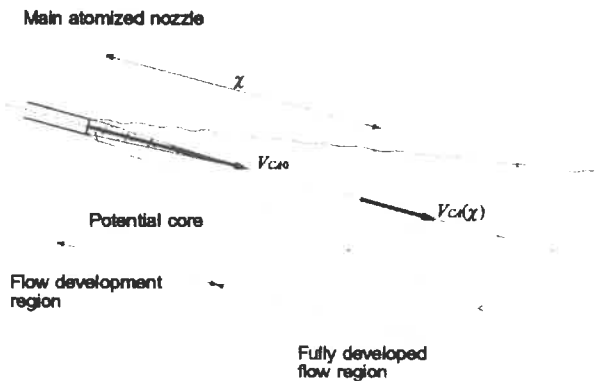


Fig.4 Model of plane turbulent free jet

that the loading current is depended on the flow velocity as follows.

First, supposing that the flow velocity V_{c0} of the atomized working fluid is equal to one of the compressed air from the main atomized nozzle, the flow velocity in neighborhood of the main atomized nozzle becomes as follows from the Bernoulli's theorem.

$$V_{c,0} = \sqrt{\frac{2(P_c - P_a)}{\rho}} \quad (1)$$

Where, P_c : absolute pressure of compressed air, P_a : atmospheric pressure, ρ : air density (760mmHg, 293K). And then, $(P_c - P_a)$ means the establishment pressure of the compressed air.

Next, when the flow of the compressed air is considered as the plane turbulent free jet as shown in Fig.4, the flow velocity on the center line of the compressed air gradually decreases after the air flow mixes heavily around the atmosphere. Where, the flow velocity $V_{ca}(\chi)$ to be χ away from the air nozzle on the center line of the compressed air can be calculated using an empirical formula by Tollmien, W., et al⁴⁾. Therefore, $V_{ca}(\chi)$ can be expressed below:

$$V_{ca}(\chi) = \frac{3.50}{\sqrt{\frac{\chi}{b_0}}} V_{c,0} \quad (2)$$

Where, b_0 means the inner radius of the air nozzle.

Fig.5 and 6 show the behavior of the flow velocity of the atomized working fluid when changing the setting point of the main atomized nozzle and the pressure of the compressed air, respectively. Where, in the case of χ less than $11.91b_0$, the flow velocity is a constant value owing to the flow development region called the potential core. According to approaching the main atomized nozzle to the machining region and increasing the pressure of the compressed air, it is thought that the loading current decreases because of increase of the flow velocity of the atomized working fluid. Therefore, the point of the main atomized nozzle is set at the zero point, but the pressure of the compressed air is 0.10MPa due to the experimental apparatus.

3.2 Cutting characteristics of changing in the total flow rate of the working fluid

The loading current and the wafer roughness are examined in the total flow rate when supplying the atomized working fluid in this paragraph. Where, the total flow rate means the sum of the flow rate of the working fluid from the main and the side atomized nozzles. For example, when the total flow rate is 40g/min and each side atomized nozzle supply 10g/min of the working fluid, the main atomized nozzle supplies 20g/min.

Fig.7 shows the relationship between the total flow rate of the working fluid and the loading current. And, these plots (●, ○) mean the loading current in the case of using the conventional supply method of the working fluid. In this figure, the loading current under each supply condition of the atomized working fluid is equal to the conventional supply method owing to 40g/min of the total flow rate of the working fluid without regard to the methods in non-vibration and vibration cutting. Furthermore, the loading current in the vibration cutting decreases than that of the non-vibration cutting under the all of the supply conditions of the working fluid. Therefore, it is clear that the vibration cutting is brought out the decreasing effect of the grinding force even if the atomized working fluid uses. Fig.8 shows the relationship between the total flow rate of the working fluid

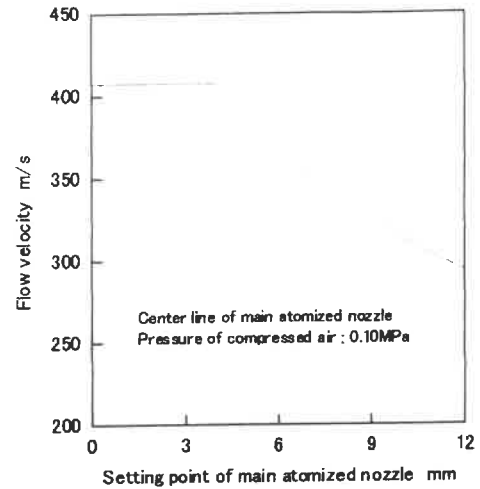


Fig.5 Relationship between setting point of main atomized nozzle and calculated flow velocity

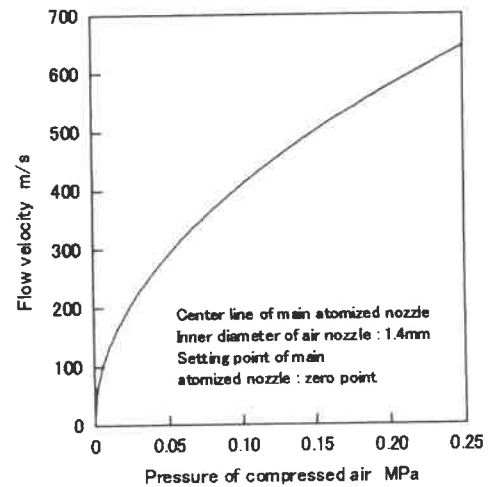


Fig.6 Relationship between pressure of compressed air and calculated flow velocity

and the wafer roughness. In this figure, it seems that the wafer roughness improves due to using 40g/min of the working fluid compared with the conventional supply method.

Next, the chipping regard with the cutting groove edge is examined experimentally. And also, the chipping amount C_p is defined and measured because of the quantitative examination of the scale of chipping. Fig.9 shows the relationship between the total flow rate of the working fluid and the chipping amount. In this figure, the chipping amount decreases as the total flow rate increases under the supply conditions of the atomized working fluid. In addition to this, it is clear that the chipping amount of the atomized working fluid is equal to one of the conventional supply method when increasing the total flow rate more than 40g/min.

4. Conclusions

As a result of executing OD-Blade vibration cutting using the atomized working fluid, the following points have been noted.

- 1)The more the flow velocity of the atomized working fluid, the cutting characteristics shows better when supplying the atomized working fluid.
- 2)The cutting characteristics supplied the atomized working fluid is equal to the conventional supply method.
- 3)The flow rate of the working fluid can be decreased by the vibration cutting using the atomized working fluid.

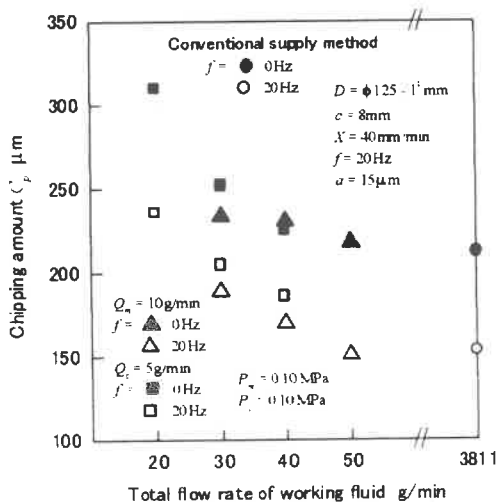


Fig.9 Relationship between total flow rate of working fluid and chipping amount

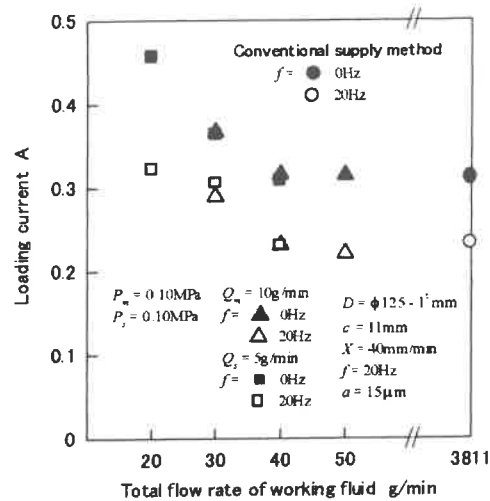


Fig.7 Relationship between total flow rate of working fluid and loading current

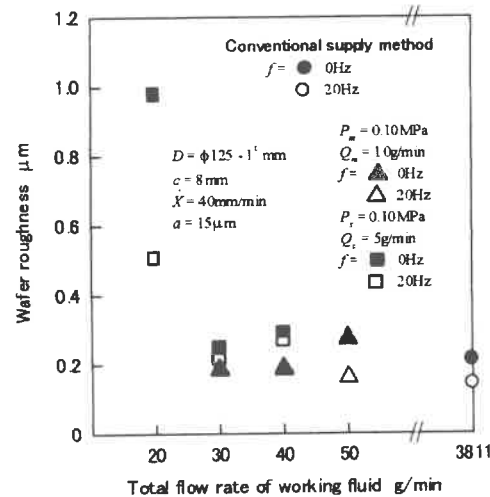


Fig.8 Relationship between total flow rate of working fluid and wafer roughness

Acknowledgments

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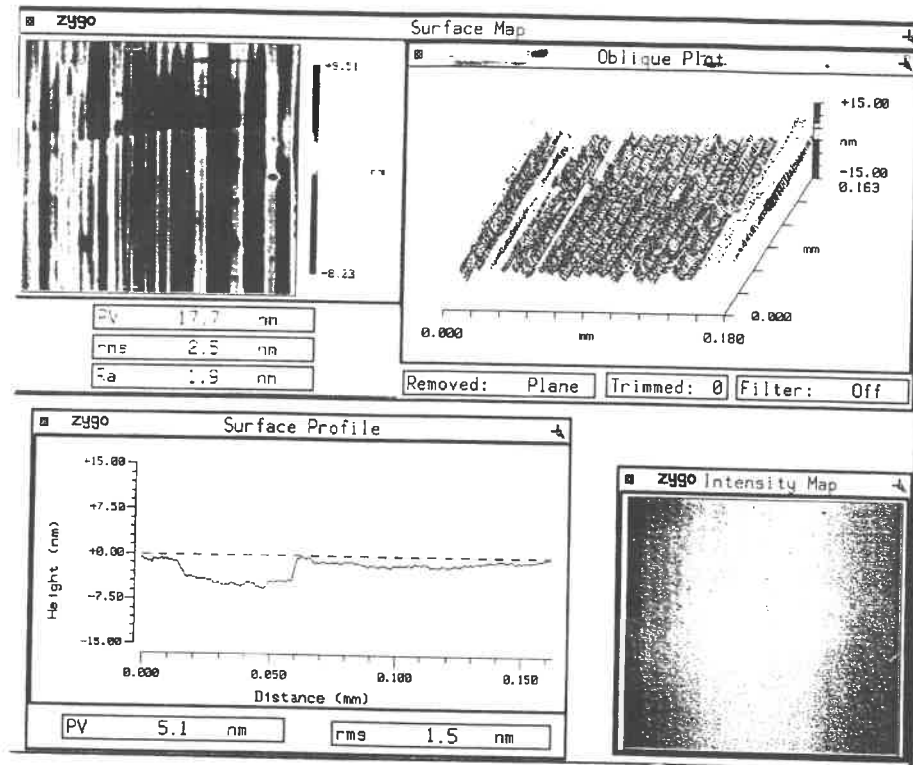


Fig.1 Ultraprecision machined surface profile ($t_1 = 2 \mu\text{m}$, $V=188.4 \text{ m/min}$)

as the bulk material. Moreover, the microvickers hardness of the ultraprecision machined surfaces were nearly the same in all cutting conditions.

The measurement results of the surface profile presented that the ultraprecision cutting provided the ultra-smooth surfaces, as might be expected, much better than the precision cutting. Fig.1 shows a typical profile of the ultraprecision machined surface. A remarkably small step, namely a grain boundary step could be observed obviously.

3.2 TEM observation

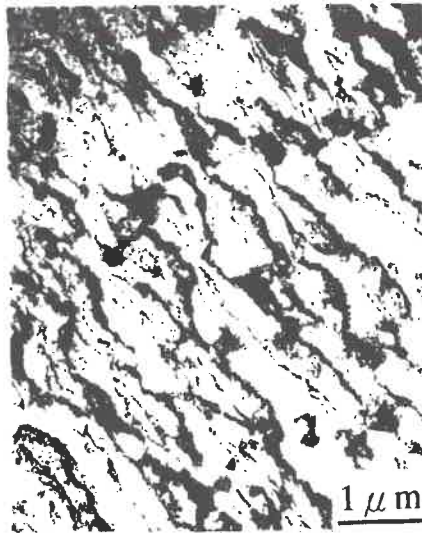
Figure 2 shows that TEM photos of precision and ultraprecision machined surface microstructure. Depth of cut $t_1 = 3, 4 \mu\text{m}$, respectively. It follows from what have been observed that there were not much differences between them on the crystallographic deformation level, namely dislocation theory. These observation results indicated that even though microvickers hardness of the precision machined surfaces were higher than the ultraprecision machined surfaces, no obvious differences in the dislocation distribution could be recognized. The reason why the precision machined surfaces had higher hardness value seem to depend on the depth of deformed layer. It must be noted that microvickers hardness value offers hardness of region being ten times as deep as indented. It should be noted that the depth of deformed layer in the ultraprecision machined surfaces were negligible small.

The precision and ultraprecision machined surfaces presented three types of dislocation structure as shown in Fig.2 and 3. One is a tangled dislocation network being similar to a chain of islands as shown in Fig.2. The other one is a cell structure as shown in Fig.3(1). Last one is an intermediate type between Fig.2 and Fig.3 (1), namely, this is somewhat uniform tangled-dislocation distribution as shown in Fig.3(2). The crystallographic orientation appears to control the type of dislocation distribution.

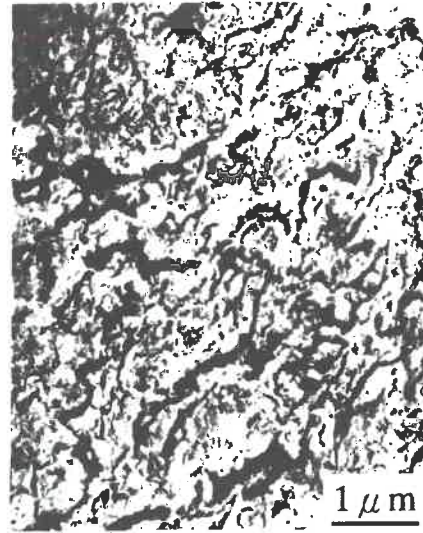
In addition, neither cutting speed nor depth of cut effects on the dislocation distribution could be observed.

4. Conclusion

All these things make it clear that the ultraprecision machined surface had much more sophisticated values than the



(1) Precision, $t_1 = 3 \mu\text{m}$, $V=25.7 \text{ m/min}$

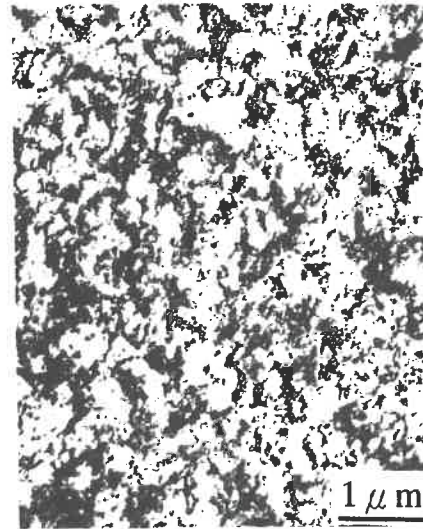


(2) Ultraprecision, $t_1 = 2 \mu\text{m}$, $V= 376.8 \text{ m/min}$

Fig. 2 TEM photo of precision and ultraprecision machined surface microstructure



(1) $t_1 = 4 \mu\text{m}$, $V=376.8 \text{ m/min}$



(2) $t_1 = 4 \mu\text{m}$, $V=62.8 \text{ m/min}$

Fig.3 TEM photo of ultraprecision machined surface microstructure

precision machined surface in the surface roughness and microvickers hardness; however, with regard to the dislocation structure, representing as a tangled network, of the machined surface, no obvious differences between them could be recognized at all. Even though the ultraprecision machined surface has remarkably thin deformed layer, the deformation at the top of the ultraprecision machined surface seem to be the same as the precision machined surface from the transmission electron microscopical view point.

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Development of a high-performance laser-guided deep-hole boring tool: optimal determination of reference origin for precise guiding

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Introduction

To bore a precise straight hole, a deep hole boring tool ought to be guided toward a target. From this point of view, the laser-guided deep-hole boring tool was developed [1-5]. The latest tool using piezoelectric actuators could be guided to go straight toward the target despite the disturbances up to a depth of 388 mm [5].

In the present paper, before the performance of the tool is examined, the following points are improved to extend the depth. Tool's guiding error, caused by misalignment of the corner cube prism and the mirror in the optical head from the spindle axis, is eliminated using a jig which determines the reference origins of the two PSDs precisely. A single-edge counter-boring head is used instead of the double-edge head used up to now. The former is thought to be better in attitude control than the latter. A new boring bar, which is 15 % lower in both, bending and torsional rigidity and which is better in controllability of tool attitude, is used.

Experimental apparatus

Figure 1 shows the experimental apparatus[5]. The counter-boring head ① has a single-edge. One cutting edge of the double-edge counter-boring head, which is used up to now, is replaced by a guide pad and six guide pads are removed [3]. The tool head consists of an optical head, a counter-boring head, piezoelectric actuators ⑫, and an actuator holder ②. The optical head ⑧ is attached to the front surface of the counter-boring head through an adjustment jig. The actuator holder is connected to a rotation stopper ⑭ behind the tool head by two parallel plates of phosphor bronze ⑥. A laser source ⑪, and PSDs ⑨, ⑩ are set in front of the tool. The optical distance between a dichroic mirror in the optical head and PSD ⑩ for measuring tool inclination is 2040 mm.

Acquisition of data for controlling of the tool

Data for tool attitude control are acquired from the two PSDs for tool position and its inclination every rotation of the counter-boring head. Until now, outputs of the two PSDs (measuring tool position and its inclination) sometimes did not correspond well to the measured hole deviation.

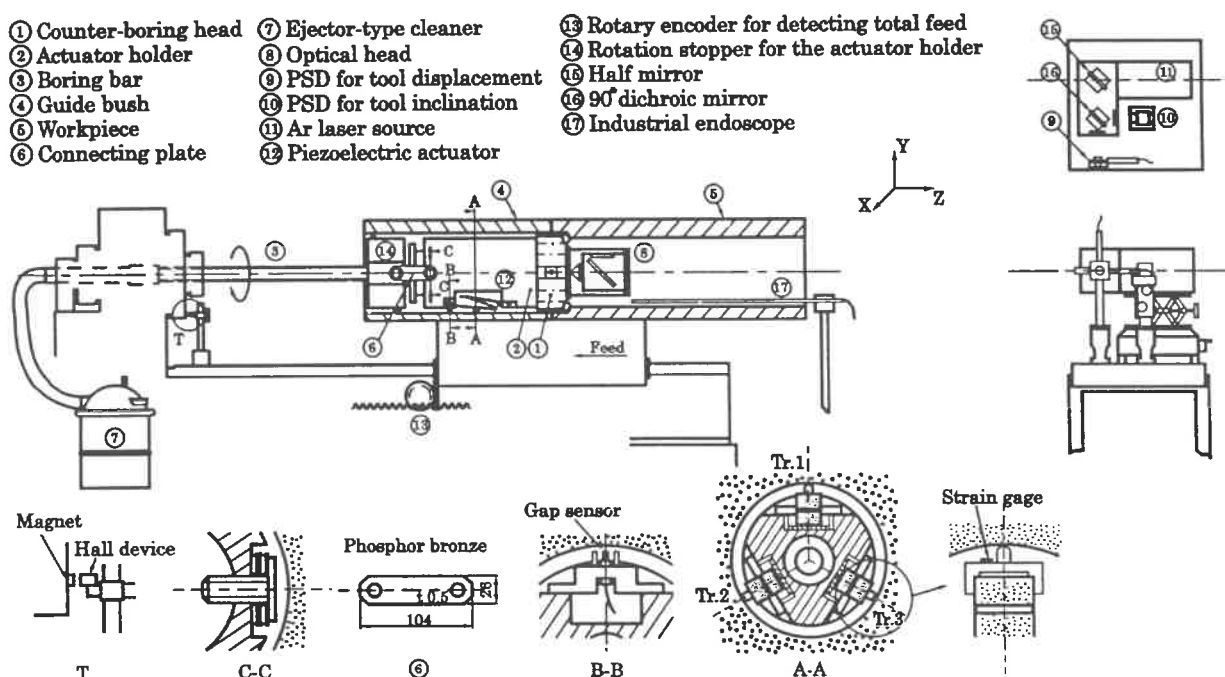


Fig.1 Experimental apparatus

To find out the cause, the following is examined. The tool head with the optical head is supported by two V-blocks and is aligned on Z-axis at the same longitudinal position as in the experiment. Then laser beam is radiated and the optical head is rotated manually.

Figure 3 shows variations of outputs of two PSDs with encoder pulse during one rotation of the optical head fixed on the counter-boring head. Theoretically, outputs of two PSDs are constant during one rotation of the optical head corresponding to 1400 pulse of output of an encoder. Changes of X- and Y- outputs of tool position are caused by change of darkness of the laser spot due to interference and polarization of the laser beam. Changes of X- and Y- outputs of tool inclination are caused by inclination of the reflecting mirror in the optical head from the Z-axis.

From the last experiment [5] on, tool position and its inclination are measured at the rotational pulse position 700 where the brightness of the two PSDs are preferable at the same time.

Misalignment of the optical parts in the optical head

Even if the laser source and the PSDs for tool position and its inclination are aligned on Z-axis, hole deviation appeared. To resolve the cause, the misalignment of the corner cube prism and inclination of reflecting mirror in the optical head from the Z-axis is examined.

Figure 4 shows all cases of alignment errors. Figure 4(a) shows that the corner cube prism and the reflecting mirror are precisely aligned on the Z-axis. Figures 4(b) and 4(c) are the cases in which the corner cube prism is displaced by ε and the reflecting mirror is inclined by θ from the Z-axis, respectively. In case of Fig.4(d), errors of Figs.4(b) and 4(c) occur together. Figure 4(e) shows the case when the optical head is inclined by γ during the setup of the counter-boring head. Figure 4(f) is the worst case, when all errors occur together. These errors cannot be eliminated by the conventional adjustment. Therefore a new guiding strategy is developed to ensure the tool can be guided straight even if the errors occur.

Optimal setup of reference origin for precise guiding

Figure 5 shows the optimal setup method of reference origins. The laser source is aligned on the Z-axis (Fig.5(a)) [5]. The optical head is fixed to the front surface of a cylindrical alignment jig through an adjustment jig.

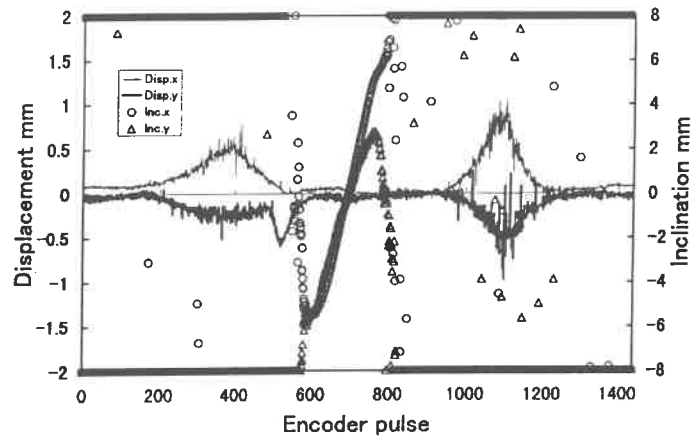


Fig.3 Variations of outputs of two PSDs with encoder pulse (during one rotation of the optical head)

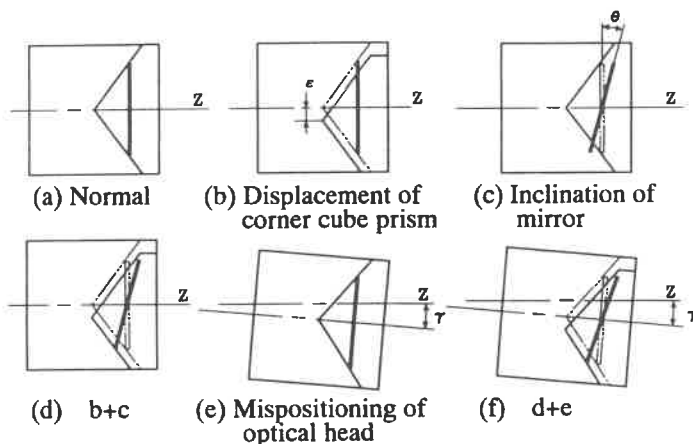


Fig.4 Misarrangement of the optical parts

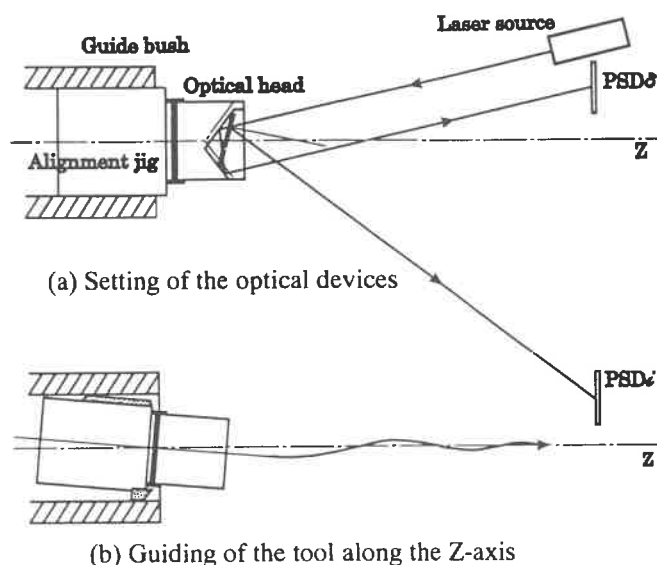
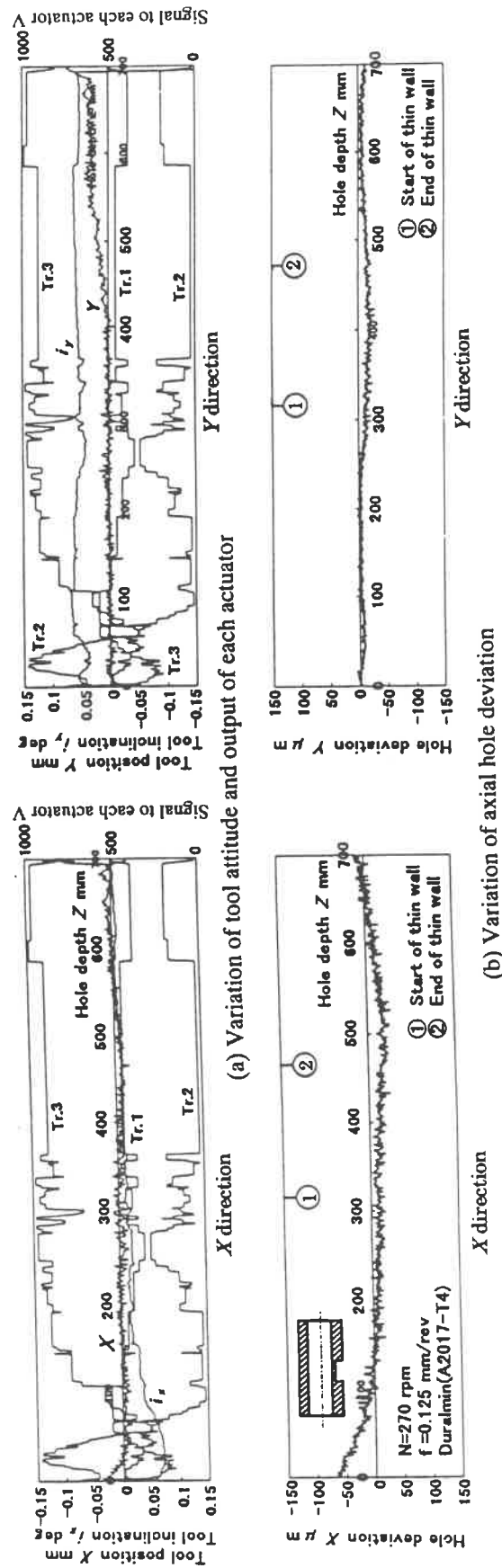
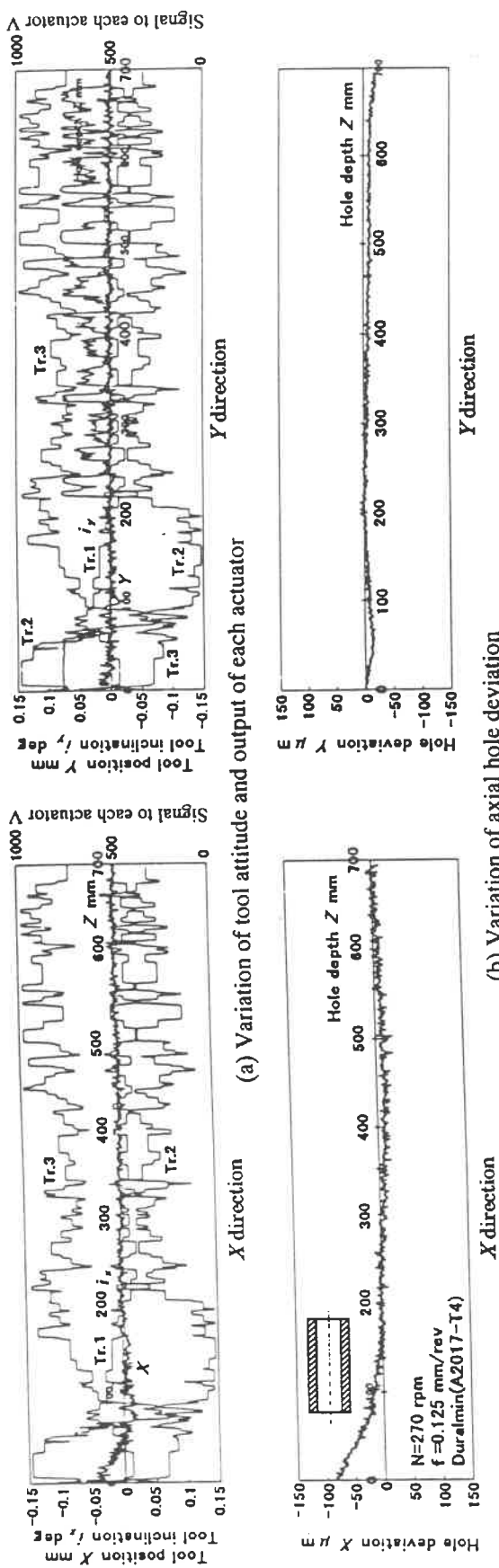


Fig.5 Determination of the origins of the two PSDs



The alignment jig is inserted into the guide bush which is fixed on a machine table and the centers of both, alignment jig and the optical head, are aligned on Z-axis. Then the laser beam is radiated. Reflected beams reach the PSDs for tool position and its inclination. When the cylinder is rotated by hand, the rotational position, at which the output is most reliable, can be found. Next, the PSDs are moved till the spots lie at their centers. This position corresponds to the pulse position 700 of the encoder. The centers are reference origins for tool position and its inclination.

At this rotational position, the optical head is fixed to the counter-boring head using the adjustment jig (Fig.5(b)). When the control starts, the tool head follows the alignment jig's axis.

Experimental method

Performance of the laser guided tool is examined using two kinds of duralumin workpieces (JIS A2017-T4) with prebored 108-mm diameter hole. The normal workpiece has no special feature in shape which causes hole deviation. The other workpiece has a thin wall (1.0 mm after boring) between hole depths $Z = 325$ and 475 mm on $+X$ side. Experiments are conducted using the rotating tool-stationary workpiece system. The rotational speed N is 270 rpm and feed f is 0.125 mm/rev. Single-edge counter-boring head is used. A new boring bar, which is lower in rigidity and better in controllability of tool attitude, is used. Tool diameter is 110 mm. Cutting fluid is sulfo-chlorinated oil (JIS type 2-13) and is supplied from the laser side.

Results and discussion

Boring of the normal workpiece Figure 6 (a) shows the variations of tool position and its inclination in X and Y directions detected by PSDs and the voltage impressed to each actuator with hole depth. Figure 6(b) shows variations of the hole deviation with hole depth. Figure 6(a) corresponds well to Fig.6(b). The tool is well guided and a straight hole is bored. The large deviation toward $-X$ direction at the entrance of the hole is due to the fact that tool displaces largely due to the unbalanced cutting force acting on the tool head when it enters the workpiece. If the tool attitude is not controlled, the tool continues to move toward the direction $-X$ and the hole deviation increases.

Boring of the thin walled workpiece Tool attitude shown in Fig.7(a) corresponds well to hole deviation shown in Fig.7(b). The hole is bored straight although a thin hole wall (thickness $t = 1.0$ mm) exists between depths of $Z = 325$ mm and 475 mm in $+X$ side of the workpiece. Up to a hole depth of $Z = 368$ mm (213 mm after the supporting pads of the piezoelectric actuators enter the bored hole), the tool can be controlled precisely. The thin walled workpiece is lowest in rigidity near a depth of 400 mm which is the longitudinal center of thin wall. When the counter-boring head comes near this center, the workpiece deforms due to the unbalanced cutting force 167 N. The deformation of hole wall results in difficulty of tool attitude control.

Conclusions

A new laser guided BTA tool using piezoelectric actuators is developed to prevent axial hole deviation. Tool's guiding error, caused by misalignment of the corner cube prism and the mirror in the optical head, is eliminated using an adjustment jig which determines the reference origins of the two PSDs precisely. A single-edge counter-boring head is used instead of the double-edge head used up to now. A new boring bar is used, which is 15 % lower in both bending and torsional rigidity and better in controllability of tool attitude. As a result, controlled boring becomes possible up to a depth of 700 mm. 700 mm is the maximum machinable length of the machine tool. The tool can be put to practical use.

Acknowledgments

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DIAMOND TURNING FIGURE CONTROL BY IN-PROCESS MEASUREMENT

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1. Introduction

Large precision mirrors for optical instruments are usually manufactured by grinding and polishing the glass substrates. For high accuracy figuring, at the last stage of the polishing, many times of measurement and correction polishing are repeated. A diamond turned metal mirror is expected to be quicker and lower cost machining as well as more flexible figuring. However, without suppressing the systematic and random errors, e.g. temperature change and disturbances, the diamond turning machine is not accurate enough to cut a large optical mirror. For the accurate mirror, therefore, an ultra-precision turning machine with a well environmental controlled facility is essential and it will lose the low cost advantage.

In the past, we developed a workpiece-referred form accuracy control system with in-process measurement¹⁾, which controls the non-repeatable and/or repeatable waviness errors mainly due to the table feed motion and internal or external disturbances. This research work, on the other hand, is to develop a new method of diamond turning control with in-process measurement to correct the figure error in real time. In-process measurement is the simultaneous measurement during the machining, and it will reduce the time of machining considerably, as well as can make the cutting more accurate. The Hartmann test is one of the well-known optical measuring methods for lens aberration or concave mirror surface figure. A preliminary experiment of concave mirror turning with figure control by in-process measurement was carried out and showed its effectiveness. Since the reference of this turning control system is the figure measurement, the machine accuracy and the environmental disturbances will primarily not affect to the final mirror figure. The real-time or in-process figure control datum was obtained by newly developed "Half-line Hartmann test" system.

2. Half-line Hartmann test

In the previous paper, we reported a modified Hartmann test method, "Fiber-Grating Hartmann Test" ²⁾, using a crossed optical-fiber-grating instead of the

traditional Hartmann screen with multiple holes. A fiber-grating sheet is composed of many short optical fibers lying side by side in contact with each other. On illumination with laser light, the fiber-grating generates diverging multiple rays of almost the same intensity in the direction of interference. The fiber-grating is positioned in the center of curvature of a concave mirror and the beams return to one point from the ideal spherical mirror. Detecting the spot image by a CCD camera at near the focal point, a personal computer analyzes the image into a mirror figure error map.

In the in-process measurement from distant position on the machine, light beams are sometimes intersected by obstacles such as a tool stage, coolant pipes, etc., and it is difficult to measure all the surface figure of cutting mirror at a time. A Half-line Hartmann test method is developed to look in only a half radial part of the rotating mirror surface using a one-dimensional or line fiber-grating which illuminates on one radial direction of the mirror surface with linear multiple light spots. Figure 1 shows the generation of full-line multi-optical-beams by the line fiber-grating. A radial section figure can be measured by taking the half-line multi-spot image reflected by the concave mirror under test with 1/10000 second exposure of CCD camera and then by digital processing the image into the height distribution along spots. The full mirror surface figure can be obtained by combining the multiple exposure data for different sections of rotating mirror. Some static and dynamic setting errors are eliminated during the processing. The reliability of the measurement is confirmed by the comparison with a Fizeau interferometer.

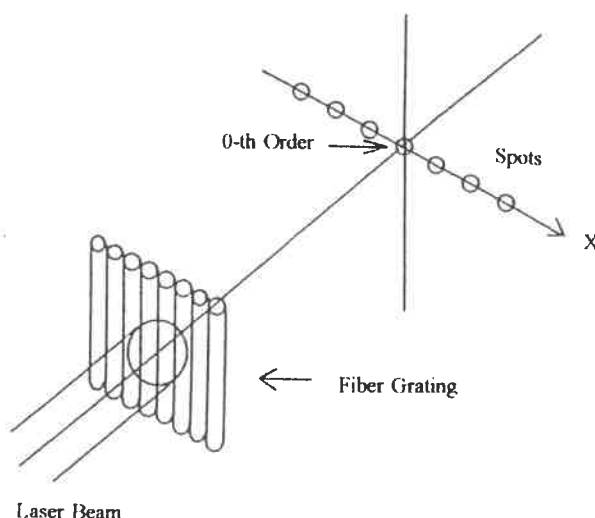


Fig. 1 Light spots from a line fiber grating

3. Figure turning control

As a preliminary experiment before the full surface control, the figure turning control is carried out for along only one radial section. The experimental set up is shown in Fig. 2. The one-dimensional multi-beam reflected from the rotating substrate under machining is detected by a CCD camera. The quasi-real time radial section

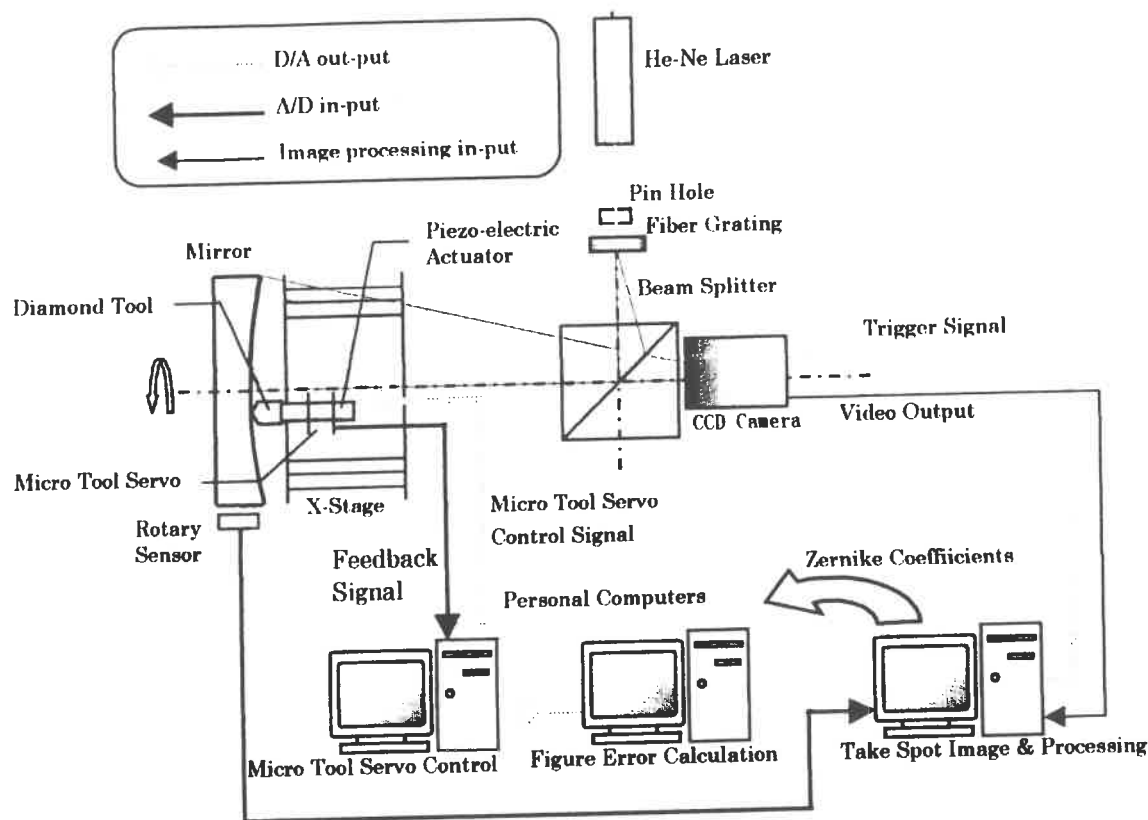


Fig. 2 In-process measurement and figure turning control experiment

datum can be obtained easily by a polar coordinate Zernike polynomial expansion of measured figure data detected by the Half-line Hartman test. The amount of control or the piezo-electric voltage to drive a diamond tool is calculated from the figure error trend of immediately previous cutting along the radial cross section. The experimental conditions are 1000 rpm spindle turning, $20 \mu\text{m}$ of depth of cut, 5 mm/min feed speed. The aluminum workpiece is 140 mm diameter with 1300 mm curvature radius.

3. Experimental results

Figure 3 shows a result of controlled cutting measured by the half-line Hartmann test in on-machine mode just after the experiment. Before the controlled cutting, the same workpiece was cut without control, of which figure error also measured by on-machine mode is shown in Fig. 4. The difference of these cross sectional figures will be the improvement with the control cutting. The peak-to-valley error value is reduced to 42% of non-controlled value.

4. Conclusion

A preliminary experiment of figure controlled cutting with in-process measurement was carried out for one cross section of workpiece surface and showed its possibility of

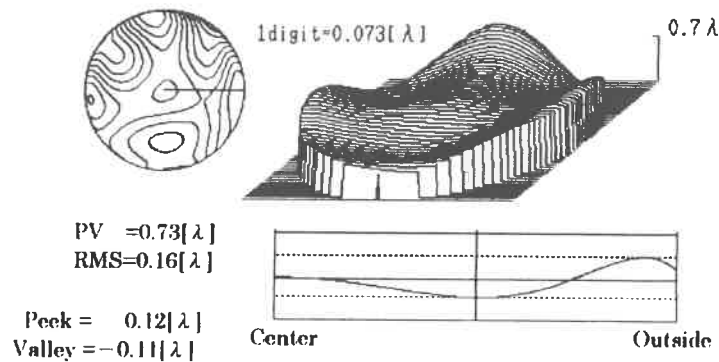


Fig.3 Result of controlled cutting

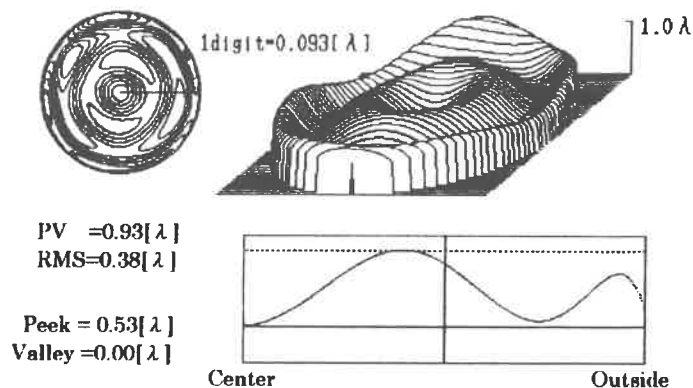


Fig.4 Surface figure by un-controlled cutting

error correction. Newly developed Half-line Hartmann test system was used as the measuring instrument. The experimental result shows clear reduction of figure error. In the near future, improving mainly for the computer software, the full surface of workpiece will be able to cut more accurately using the same measurement and control hardware systems.

5. Acknowledgement

This research work is performed in connection with a research group of JSPE "In-process measurement and machining control". The authors are indebted to the members, especially to Dr. Shigeru Ueno of the Technical Research Institute, Jan. Soc. Prom. Mach. Ind., for his special design and manufacturing the turning machine, as well as to Dr. Yutaka Uda of Nikon Co. for useful discussions.

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SURFACE ROUGHNESS VALUATION OF SI WAFERS BY DIAMOND TURNING

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Introduction

The TTV (Total Thickness Variation)¹⁾ value of silicon wafers becomes more important in the fabrication of LSI. For 200 mm wafer, the TTV value should be less than the 0.1 μm . To obtain smaller TTV value for 300 mm or 400 mm wafers, the depth of the damage at each slicing and lapping process should be reduced. In the previous work²⁾, two types of the precision grinder and the single point diamond (SPD) turning were applied to wafering process and surface roughness were measured using cross-sectional TEM. Obtained results were compared using Discrete Fourier Transformation (DFT) for characterization of surface roughness. The DFT characterization was found to be a useful method to detect both short range and long range waviness of the surface.

In this work, the surface roughness and the damage depth were measured in sub-nanometer scale. The surface roughness of a Si wafer by SPD turning was valued using cross-sectional TEM images. The TTV values were measured in nm scale from high resolution micrographes. LSI fabrications are now proceeding to 0.2 μm line rule, further integration requires better TTV values. Introducing the diamond turning at the lapping process will reduce the final mechanical chemical etching depth and improving the TTV value. The wafer surfaces by SDP turning have smaller roughness values than by conventional precision grinding, precise valuation methods such as by AFM or by Cross-sectional TEM were required.

Experimental

A Si 100 wafer of 200 mm diameter has been carried out by using a single point diamond turning with various feeding rate and cutting conditions. Pure water was used as lubricant. The wafer was finished by 7 different conditions which were listed in Table 1. Then the wafer was broken into three pieces, for TEM, AFM and TWIN inspections. Schematic illustration is shown in Fig. 1. The wafer surface of No. 2, 3 and No. 6 area were observed and discussed in the present work.

TEM observations were performed using a Hitachi H-8100 microscope operated at 200 kV. Cross-sectional specimens were prepared by conventional methods including mechanical thinning with a dimple grinder and 5 kV Ar ion milling.

No	feed rate [$\mu\text{m}/\text{turn}$]	cutting speed [RPM]
1	10	1,308
2	10	1,650
3	10	1,700
4	10	1,800
5	14	1,900
6	14	2,000
7	13	2,200

Table 1

Table 1 Feeding rates and cutting speeds of the SPD turning are listed.

Results and discussion

Fig. 2 shows the micrograph of the cross-section of No. 3 area of the wafer. The feed rate was $10\text{ }\mu\text{m}/\text{turn}$ and cutting speed was 1,700 RPM. Incident electron beam direction was perpendicular to 110 plane. As can be see from Fig. 1, the turning trace circles on the wafer, the obtained cross-section is not exactly perpendicular to the cutting direction. The inset of Fig. 2

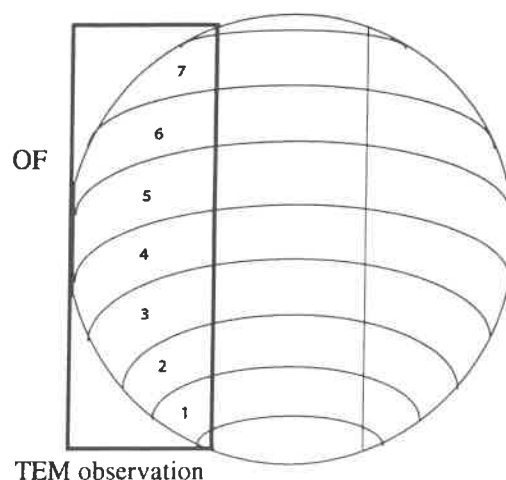


Fig.1

Fig. 1 Schematic illustration of the 200 mm Si wafer, finished by various feeding rate and cutting speed of single point diamond turning.

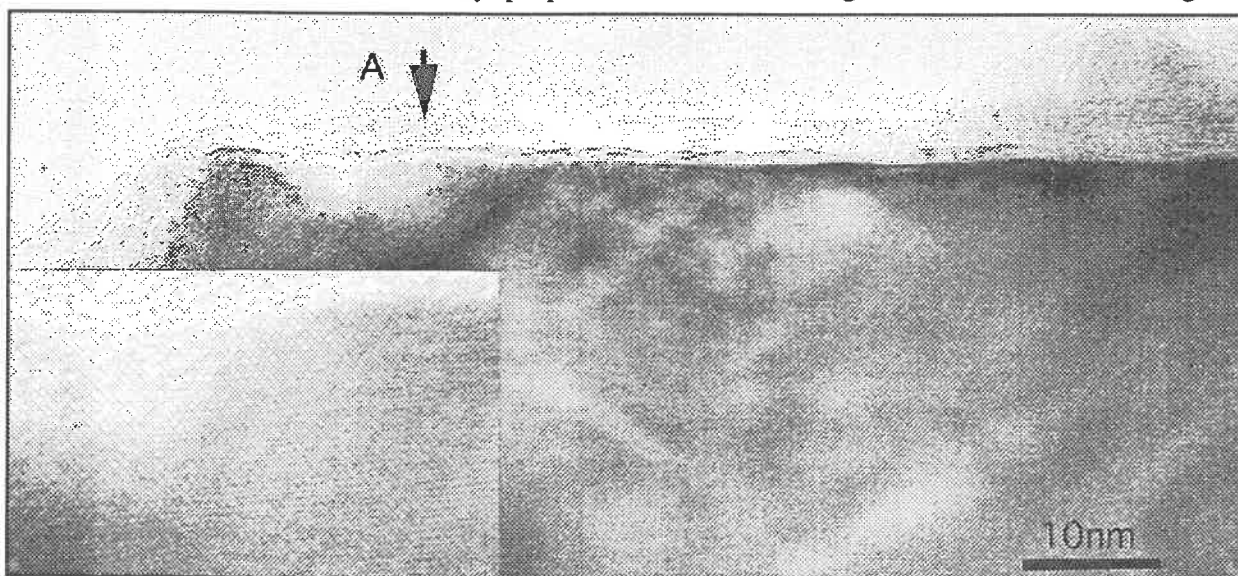


Fig. 2 Cross-sectional TEM image of the No.3 specimen. Inset is Enlarged image of A part (arrow). 111 and 200 lattice fringes can be seen. Incident electron beam was perpendicular to 110 plane.

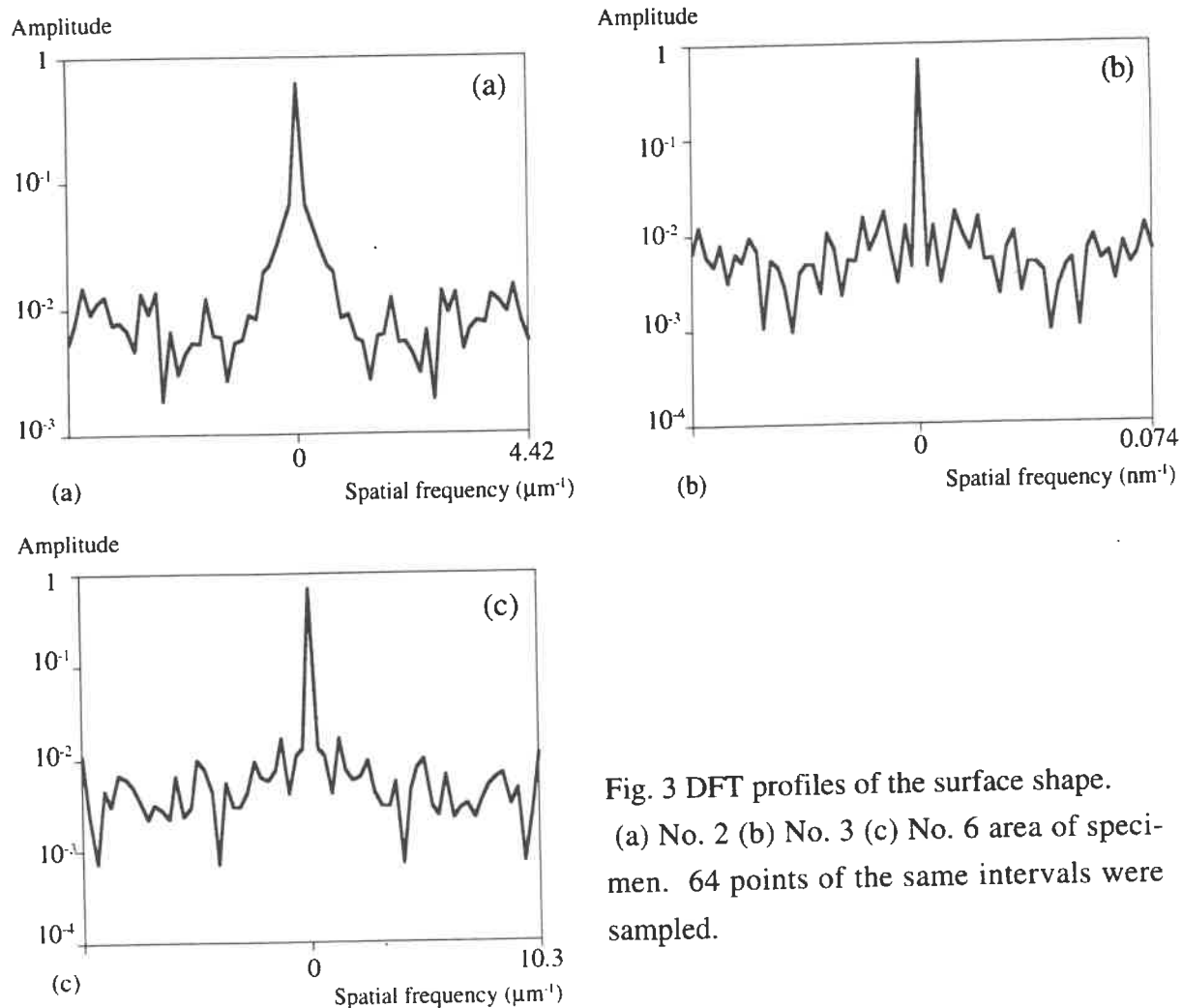


Fig. 3 DFT profiles of the surface shape.
 (a) No. 2 (b) No. 3 (c) No. 6 area of specimen. 64 points of the same intervals were sampled.

is a enlargement of the surface, in which the 0.32 nm lattice fringes can be seen. The lattice distance is corresponding to 111 plane which are parallel to the surface. Another fringes have 0.27 nm distance which corresponds to 111 planes. The roughness can be measured in accuracy of 0.32 nm by tracing the surface steps.

DFT profiles are shown in Fig. 3 (a), (b) and (c). The height of the 64 points of same interval distance were measured from a certain line drawn parallel from the lowest point of the surface. The x-axis of the profiles show the spatial frequency of the waviness. The y-axis show the population of the waviness of each frequency. As the magnification of the cross-sectional micrographs were different, the x-axis of each figures have different frequency value. The amorphous layer of 10 nm thickness on the surface was observed, this is the native oxide layer and no amorphous layer which has the mechanical origin was observed. The roughness values Rz (ten point height) and damage depth were measured from high resolution cross-sectional TEM images of No. 2, No. 3 and No. 6 part of the wafer. The result are listed in Table 2. The obtained

RPM	Rz	Damage Depth
1,650	51.7nm	366nm
1,700	3.08nm	22nm
2,000	7.1nm	NA

Table 2 Surface roughness value (Rz) and the damage depth are listed.

damage depth values are also listed. For example, the value of 366 nm (No. 2 wafer) means the dislocation length from the surface of the mechanically finish surface. Surface roughness value (Rz) for example was 51.7 nm. This value is smaller than the previous diamond turning result. Obtained values are compared with AFM result and show good agreement. Rz value of No. 3 is smaller than that of No. 2 while the feeding rate is same and the cutting speed are 1,650 and 1,700 RPM. This is because the magnification of the micrograph has x500,000 that is 10 times larger than the other two, the measured area of No. 2 was small.

Summary

Surface roughness and damage depth of a Si wafer which was finished by various feeding rate and cutting speed of SPD turning were measured by using high resolution cross-sectional TEM images. The DFT profiles of the surface shape show surfaces have lower waviness in wide spatial frequency area.

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Fast Tool Servo System and its Application to Three Dimensional Fine Surface Figures

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1. Outline

A fast tool servo system has been developed to apply three-dimensional fine surface generation by diamond turning. Two types of fast tool servo mechanisms have been developed for face turning and boring operations. Each utilizes a piezoelectric actuator, a built-in capacitance type displacement sensor and an analog circuit for closed-loop control. A computer and digital-to-analog converter command the mechanisms by feeding the servo a series of displacement data along the tool path of the workpiece surface. This displacement data is synchronized to the spindle rotation. Several sample pieces have been machined as examples, including some types of segment mirrors, function-defined surface forms and bitmap-defined figures. Patterning on internal cylindrical surfaces down to 40mm in diameter has also been done, indicating an application to journal fluid bearings.

2. System configuration

The essence of the system is that the depth-of-cut along the tool path is controlled at a very high update rate, synchronizing to the spindle rotation. A fast tool servo ^{[1][2][3]} is a key device to control the depth-of-cut with high accuracy and quick response. Some modified commercial lathes already exist and that can manufacture non-axisymmetric objects. However, the obvious difference between the preceding technology and the system reported here would be that in this system the depth-of-cut control does not repeat and that the update rate is rather high, leading to higher spatial resolution. Very similar research has been also done in North Carolina State University ^{[4][5][6]} and TH Aachen ^{[7][8]}, Germany.

The system consists of an ultra-precision diamond turning machine, a fast tool servo mounted on the tool slide of the machine, and a controller based on a PC (personal computer). The fast tool servo is driven by a piezoelectric actuator, together with compensatory analog circuitry for closed loop control. The rotation of the

lathe's spindle of the lathe is detected by a rotary encoder with 4096/rev resolution, and fed as an updating pulse to the PC via a DMA (direct memory access) capable digital-to-analog converter. The numerical controller of the lathe directs only the fundamental motion of the spindle and the tool slide, independent of the fast tool servo, but the position signal is fed to the

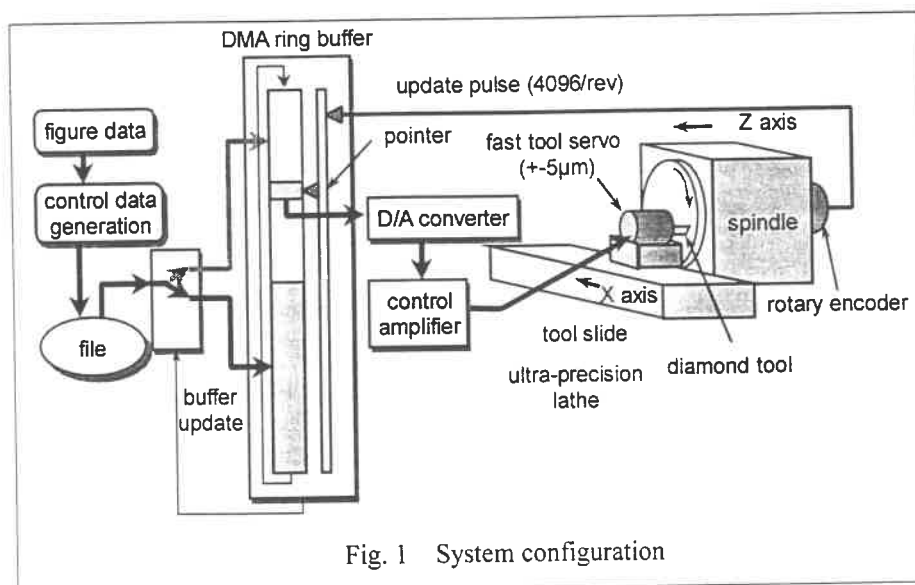


Fig. 1 System configuration

PC.

3. Generation of cutting control data

Cutting control commands are generated as a string of depth-of-cut data along the tool path for the workpiece. The data is first generated from the definition of the surface figure to be machined, and then fed to the DMA ring buffer, preventing from any disturbance to DAC update. The surface figure is defined in several formats including bitmap definition, exact description by mathematical functions and a group of facets. These formats are converted into the control data by the individual converting program, and stored as a file.

In reality, the resolution of the surface figures are restricted by many parameters, e.g. resolution of the rotary encoder, the nose radius of the tool, the track pitch of the tool path derived from the feed rate and the spindle rotation, and the response bandwidth of the fast tool servo. These parameters must be compromised to achieve acceptable surface roughness, machining duration and the control data size. To obtain a finer resolution, a sharper tool chip, finer track pitch, slower spindle rotation, faster tool servo, and a higher encoder resolution are required. Moreover, the accuracy of the surface depends on the basic accuracy and stability of the machine.

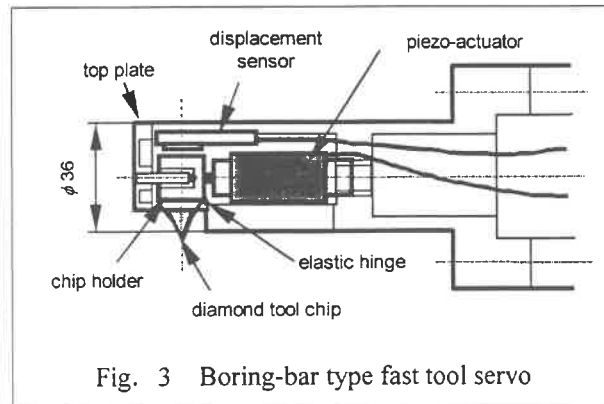
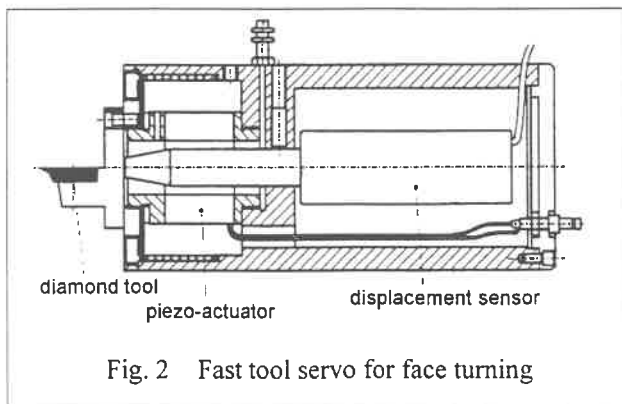
4. Fast tool servo mechanisms

4.1 Face turning fast tool servo

The use of a piezoelectric actuator in a fast tool servo is a common way. A stacking ring piezoelectric actuator of 25 mm OD x 14mm ID x 19mm long is fixed at one end inside a stainless steel cylindrical, sizing 53mm in diameter and 112mm long (fig. 2). The moving end is supported by a diaphragm to suppress excess displacement modes, and a tool holder carrying a diamond tool chip is attached to it. The displacement of the moving end in reference to the housing is directly measured by a capacitance type displacement sensor. The actuator, flexure support, displacement sensor and the tool chip are aligned on the same centerline. The maximum displacement is about 15 μm for an applied voltage of 600 V. The primary resonance appears at 10 kHz. Utilizing a conventional single closed loop control with a notch filter component, a flat bandwidth of 2.5 kHz without overshoot and effective travel of 10 μm was achieved, together with improved stiffness.

4.2 Boring-bar type fast tool servo

An alternative fast tool servo was developed and tested to machine the internal surface of a cylinder (Fig. 3). A piezoelectric actuator is axially mounted in a boring bar, and pushes the chip holder block against the top plate. The displacement direction is diffracted by 90 degrees using a hinge mechanism. The displacement sensor is required for operation, and is installed facing the chip holder. The first resonance occurs at 8.5 kHz. In the same manner as the former FTS, a bandwidth of 2.5 kHz and 8 μm are obtained. The minimum inner diameter possible for machining is 40 mm.



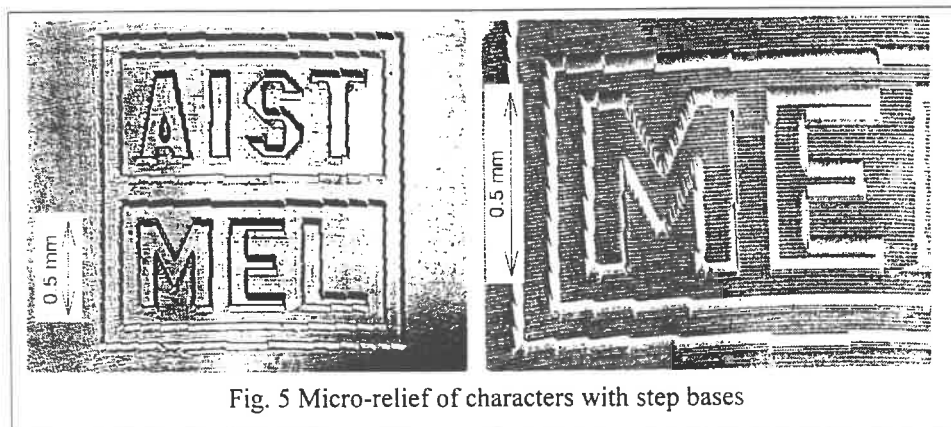
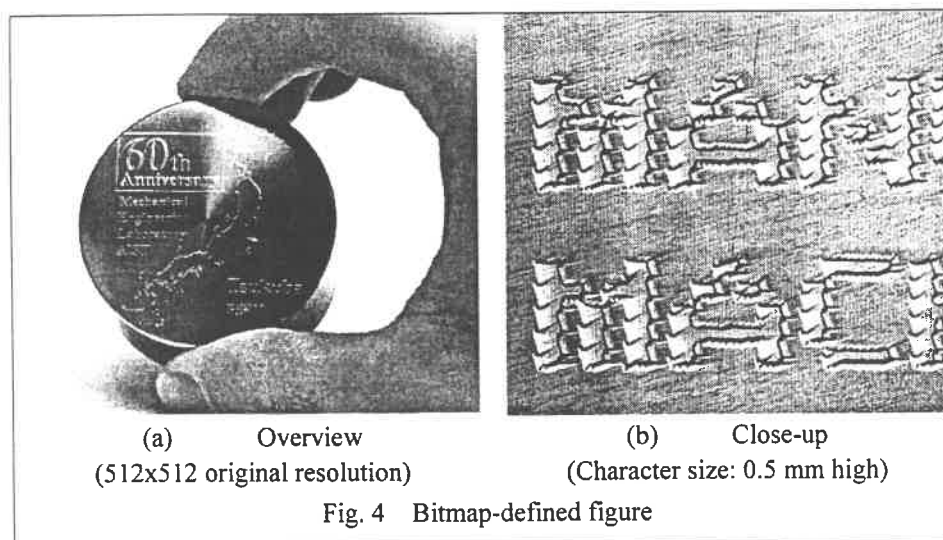
5. Machined examples

Machining tests were performed on several materials, e.g. aluminum alloy, oxygen-free copper, thick plated electroless nickel and PMMA. Terrapin oil mist spray was used as coolant/ lubricant for the metal workpieces. The tool nose radius used was 0.2 mm to 1.2 mm. The spindle rotation was set as slow as possible, 89 rpm, in order to maintain the response of the FTS. The feed rate was 1 to 2 mm/min resulting in the track pitch of 11 to 22 μm . The base diamond turning machine used was the Toyoda AHN 30x25 which features hydrostatic spindle and slides, and a 10 nm laser feedback system. The machining duration was calculated to be as short as 15 to 30 min.

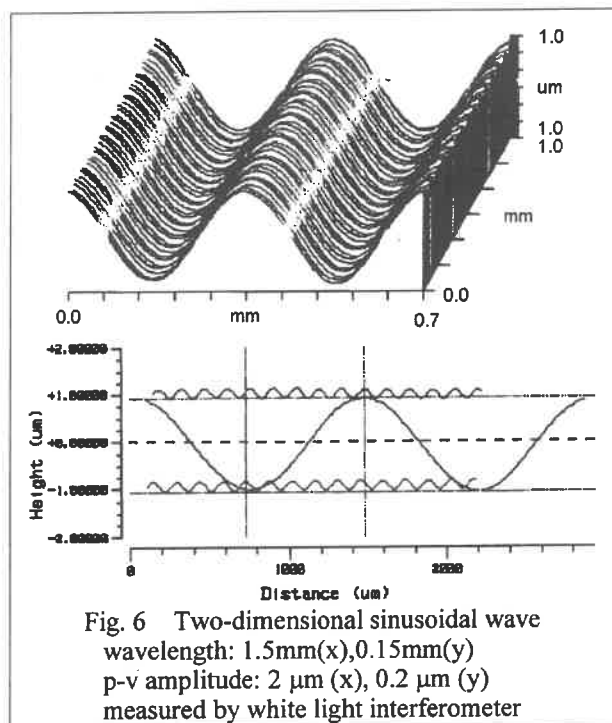
Fig. 4 shows a machined sample of a figure defined by a 512x512 bitmap data. The color in the data represents height information. A close-up (b) shows the resolution of the original design.

Fig. 5 shows a micro-relief of 0.5-mm high characters. Each step is 2 μm in height. The edge is finer than in Fig. 5(b). The rippling in the vertical line is caused by the limited angular resolution, and that in the horizontal lines is due to the infinite track pitch.

Fig. 6 shows an example of the machined surface defined by a mathematical function, of two-dimensional sinusoidal waves. A sinusoidal wave of as fine as 0.2 μm of amplitude in y-direction is machined. This sample is for a novel position-detecting device. Other figures such as a set of facets, segmented mirrors and micro-step mirrors with high accuracy were machined through face turning. Fig. 7 is a photograph of a "makyō"; a device which projects an image when illuminated by a parallel light beam, due to slight dimples (around one μm) on the mirror surface.



The boring-bar-type fast tool servo generated several on the internal surface of cylinders. A world map defined as a 1024x530-bitmap data was successfully machined on an internal cylindrical surface of a transparent PMMA tube. Fig. 8 is a close-up of the surface containing "Philippine islands", observed through the plastic tube. The pixel size is 123x75 μm . The possible applications include pockets of hydrostatic bearings and herringbone grooved aero/hydrodynamic bearings.

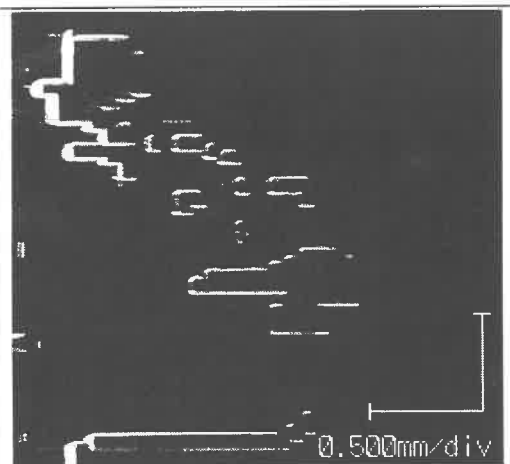


6. Conclusion

The agile machining method presented has the potential to manufacture three-dimensional fine surface figures with high precision. Possible applications include mechanical instrumentation standards, non-conventional optics, bearing components, boundary layer control surface, electronics and printing.

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INFLUENCE OF COOLANT TEMPERATURE ON SURFACE FINISH IN TURNING OPERATION

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Abstract *The paper deals with experimental studies to determine the influence of coolant temperature on surface finish. Turning operation was performed on different work and tool material combinations and at different coolant temperatures. It was observed that there is an improvement in the surface finish due to decrease in the temperature of coolant. The details of the experimentation, results and discussion are presented in this paper.*

1.0 Introduction

In reality it is not possible to produce perfectly smooth surface. The machined surface always departs from absolute perfection to some extent. The imperfections on the surface are in the form of succession of hills and valleys varying both in height and spacing. The surface texture is influenced by the machining process employed, type of coolant and its temperature, tool material, work material etc. In particular, the coolant temperature helps to keep the tool cool and prevent it from being heated to a temperature at which the hardness and resistance to abrasion are reduced. Also, coolant helps to keep the work cool, thus preventing it from being machined in a warped shape to inaccurate final dimensions.

The precision machining is becoming much popular in recent years for machining of precision parts such as aerostatic bearing, aerodynamic gyrobearing, ultra precision X-Y tables, etc. The machining accuracy in precision machining is mainly predominated by the thermal deformation of cutting tool as well as the accuracy of machine tool itself. Many researches have been carried out to study the influence of coolant temperature on metallurgical aspects as well as machining aspects. The temperature rise of the tool and the workpiece due to the cutting heat is not negligible even in case of micromachining and its effect on the machining accuracy should be taken into consideration especially when higher machining accuracy is required [1-3].

In this paper the experimental analysis of the influence of coolant temperature on the surface finish of the workpiece in turning operation is presented and discussed. This study has potential for extending the investigations to ultra precision machining which is expected to produce high accuracy components.

2.0 Influence of coolant temperature on machining

It is well known that most of the workdone in metal cutting is converted into heat, resulting in a temperature rise in the tool, workpiece and chip. The effect of cutting temperature on hardness of tool material is shown in Fig. 1. The rise in the temperature results in decrease in tool hardness due to which nose radius is effected affecting the surface finish [4].

To lower the cutting temperature chilled coolant is used. The coolant is applied locally at the cutting tool tip and workpiece interface. With the development of difficult to machine materials, the tool tip temperature control may become advantageous. The present work explores the benefits of considering the tool temperature as a machining variable. The heat generation due to machining, between tool and work, can be varied by varying the coolant temperature resulting in better surface finish.

3.0 Experimental procedure

The schematic diagram of the experimental setup is shown in Fig.2. The operation employed was turning. The two possible ways of supplying coolants during the turning operation are:

a) Gravity feed method, and b) Pump feed method

Pump feed method was adopted for supply of coolant. A coolant tank of 10 liters capacity which was properly insulated with husk and thermocoal was used. A thermometer was used to measure coolant temperature. Single point H.S.S and carbide tipped tools were used in the investigation. Some details of the setup are given in Table 1.

Table 1. Experimentation details

a) Lathe used for experiment: EP 1330 manufactured by Mysore Kirloskar Ltd.
b) Spindle speed: 315 rpm
c) Diameter of workpiece: 30 mm
d) Flow rate of coolant: 3 LPM
e) Type of coolant: Soluble oil
f) Nose radius: 0.5 mm
g) Room temperature: 24 °C

Workpieces of two different compositions were turned. Coolant at different temperatures was applied during cutting. The workpiece was held in a precision three jaw chuck. Automatic feed was given and the workpiece was turned at different feed rates and depths of cut. After turning operation, surface finish of the workpiece was measured using precision instruments (Mecrin-3 and Tallyrond surface tester). Same tool nose radius was kept for all the cutting tools used in the experimentation. The radius was measured using an optical profile projector. Keeping all the parameters same, the workpiece was turned applying coolant at 15, 18 and 24 °C. Coolant was chilled using ice chamber kept near the test rig [5].

4.0 Results and discussion

Fig. 3 shows the variation of surface roughness with coolant temperature at depths of cut of 0.25 mm, 0.20 mm and 0.10 mm. The feed rate was 0.045 mm/rev, tool material was tungsten carbide and work material was EN-9. In all the three cases the surface roughness has decreased by 0.1 μ m with lowering of coolant temperature.

Fig. 4 shows variation of surface roughness with coolant temperature at depths of cut of 0.5 mm and 1.0 mm. The feed rate was 0.09 mm/rev, tool material was HSS and work material was mild steel. In both the cases the reduction in roughness was 0.1 μ m with lowering of coolant temperature.

Fig. 5 shows the variation of surface roughness with coolant temperature at depths of cut of 0.5 mm, 1 mm and 1.5 mm. The feed rate was 0.045 mm/rev, tool material was HSS and work material was mild steel. At the depth of cut of 0.5 mm the reduction of surface roughness was 2 μm where as at the depths of cut of 1.0 mm and 1.5 mm the reduction of roughness was only 0.1 μm with lowering of coolant temperature.

Fig. 6 shows the roughness profiles of the workpiece measured using Tallyrond surface tester. The workpiece was EN-9 and the tool material was tungsten carbide. The experiment was conducted without coolant and with coolant at different temperatures. The figure clearly shows the improvement in surface finish with coolant at lower temperatures.

5.0 Conclusion

Influence of coolant temperature on surface finish was studied. A compact experimental set up was designed and fabricated. Experiments were conducted at various parameters like different coolant temperatures, work and tool material combinations and different depth of cut. Studies have indicated that there is moderate improvement in the surface finish due to the lowering of the coolant temperature. These studies can be extended to ultra precision machining processes to study the influence of chilled coolants on surface finish.

6.0 References

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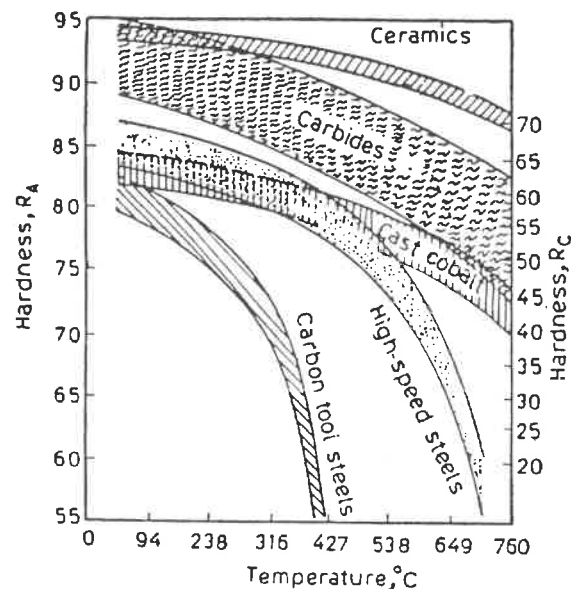


Fig. 1. Influence of cutting temperature on hot hardness of several types of tool [4]

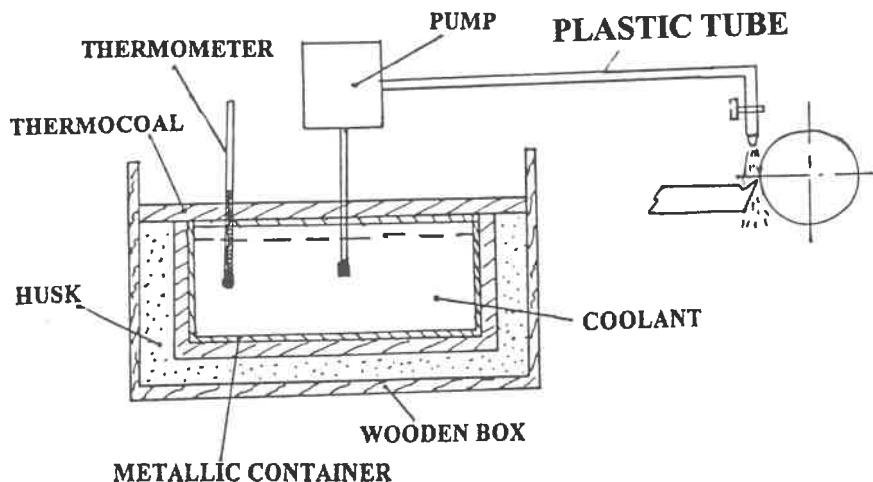


Fig. 2. Schematic diagram of the experimental setup

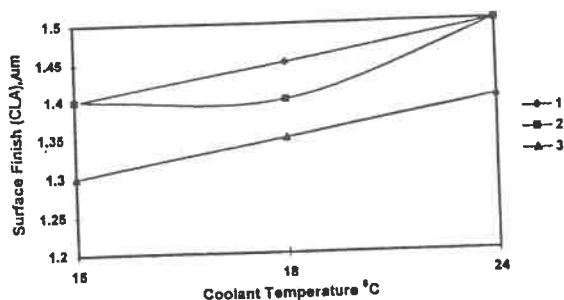


Fig. 3. Effect of coolant temperature on surface finish
Tool : Tungsten carbide ; Work : EN9 ;
Feed rate : 0.045 mm/rev
1 : Depth of cut = 0.25 mm
2 : Depth of cut = 0.20 mm
3 : Depth of cut = 0.10 mm

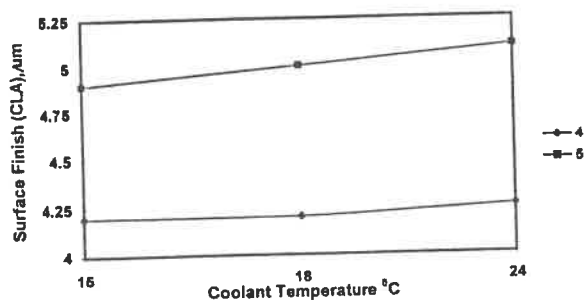


Fig. 4. Effect of coolant temperature on surface finish
Tool : High Speed Steel ; Work : Mild Steel ;
Feed rate : 0.09 mm/rev
4 : Depth of cut = 0.5 mm
5 : Depth of cut = 1.0 mm

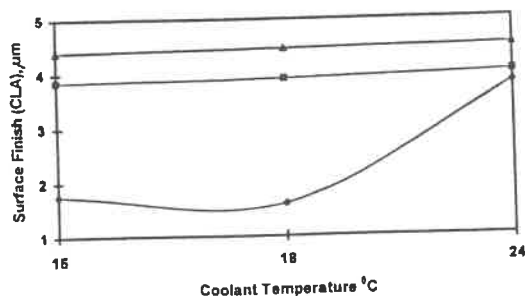


Fig. 5. Effect of coolant temperature on surface finish
Tool : High Speed Steel ; Work : Mild Steel ;
Feed rate : 0.045 mm/rev
1 : Depth of cut = 0.5 mm
2 : Depth of cut = 1.0 mm
3 : Depth of cut = 1.5 mm

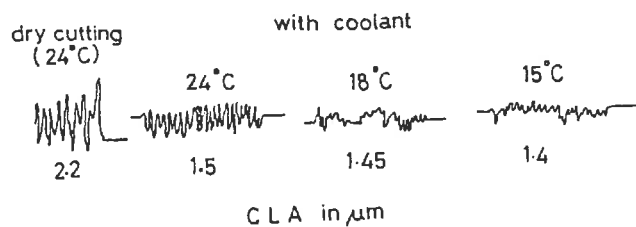


Fig. 6. Surface roughness profiles
Tool material : Tungsten Carbide
Work material : EN9
Feed rate : 0.045 mm/rev
Depth of cut : 0.25 mm

MICROSTRUCTURE GROOVES LESS THAN 50 μm WIDE CUT WITH GROUND HARD METAL MICRO END MILLS

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Introduction

Among the microstructuring techniques, such as optical or X-ray lithography, etching or laser ablation, there is a growing potential of mechanical microengineering methods. The classical precision mechanics have been developed towards higher lateral structural accuracy and better surface quality. In the past, the researchers' interest focused on machine development, resulting in ultra-precision machines with a positioning accuracy in the nanometer range for the cutting of ultra-precise geometries and surfaces /1/, on the development of methods for the cutting of so far "impossible" material-tool combinations, e.g. cutting steel with diamond tools /2/ as well as on the development of tools, cutting and mounting strategies and the combination of mechanical microengineering methods with other structuring techniques. Within the framework of the latter activities, high-precision mechanics have been developed at Karlsruhe Research Center for the manufacturing of microstructured prototypes, microsystem components /3,4,5/, molding tools /6/, micro heat exchangers, micromixers and microreactors for applications in chemical engineering /7/.

The basic techniques are milling and turning on conventional CNC high-precision machines with customarily shaped microtools. Profiled microdiamonds, e.g. of rectangular or triangular shape, with a minimum cutting edge width of below 50 μm are used to cut linear grooves into aluminum, copper, brass and other non-ferrous materials. Applying two sets of cuts in perpendicular directions results in microprisms and micropyramids, respectively. Ferrous materials, in particular stainless steel, can be cut using profiled hard metal tools. For non-linear grooves, the structural dimensions are limited by the size of the end mills available. Currently single-edge diamond end mills with a minimum cutting diameter of 300 μm /8/ and hard metal end mills with a minimum diameter of 150 μm exist /9/. Our aim was to reduce these size limitations by developing hard metal microtools with cutting diameters less than 50 μm .

Methods

Tool manufacture

Starting from the grinding techniques using diamond grinding disks as applied for the commercial manufacture of hard metal tools, we used a EWAG W11 grinding machine equipped with a set of diamond grinding disks and K10 fine-grained hard metal rods as tool material. The microtools were shaped as single-edge end mills. Due to the brittleness of the hard metal, a certain percentage of the tools broke during the grinding process, depending on the tool diameter and length, respectively.

Cutting parameters for conventional tools and microtools

The workpiece materials used for the test cuts are brass (CuZn 40 Pb 2) and stainless steel (X 5 CrNi 18 10). The cutting parameters for milling using hard metal end mills /9/ are as follows:

Material:	X 5 CrNi 18 10	CuZn 40 Pb 2
v_c	60...120 m/min	150...300 m/min
n for cutter diam. 5 mm (standard)	3 800...7 600 1/min	9 500...19 000 1/min
for cutter diam. 0.5 mm (extrapolated)	38 000...76 000 1/min	95 000...190 000 1/min
for cutter diam. 0.05 mm (extrapolated)	380 000...760 000 1/min	950 000...1 900 000 1/min

v_c : cutting speed

n: number of revolutions, calculated from the standard cutting speed

When using micro end mills, the number of revolutions theoretically required to achieve the recommended cutting speed is far above the technical limit of the available (high-speed) spindles. Thus, the achievable cutting speed will be far below the common values, e.g. $v_c = 6 \text{ m/min}$ for $n = 40\,000 \text{ 1/min}$ of a diam. $50 \mu\text{m}$ tool. Much more important is the appropriate choice of the feed and depth of the cut in order to achieve reliable cutting of high aspect ratios and satisfactory cutting lengths. Reasonable values are $\leq 10 \mu\text{m/tooth}$ feed and $\leq 20 \mu\text{m}$ depth of the cut. Lubrication and cooling was ensured by spraying of oil (Accu-Lube LB 2000).

Test structures

Structuring was done on a KERN 22/16 high-precision CNC milling machine with a spindle for max. $20\,000 \text{ 1/min}$. Most tests were performed in brass in order to optimize the cutting parameters and to determine the minimum tool diameter and groove width, the maximum aspect ratio and the cutting length. Additional first tests were performed in stainless steel. The test structures consisted in linear and curved grooves which were cut with rectangular or slightly V-shaped tools. Each groove was cut in several steps according to the depth of the cut, and vertical feed of the tool in the workpiece was avoided to minimize the tool stress.

Results

Hard metal micro end mills could be ground with a minimum diameter of less than $50 \mu\text{m}$ (Figure 1). Due to the risk of tool breaking, their length should be kept to a minimum. Mills with an aspect ratio of 1-2 are very robust, but aspect ratios of 5-10 are realizable. Microstructured hard metal microtools open up new structural dimensions for mechanical microengineering. The structuring tests showed that grooves with a width of less than $50 \mu\text{m}$ could be cut into brass (Figure 2) and grooves with a width of about $100 \mu\text{m}$ into stainless steel. The structuring depth could exceed an aspect ratio of 3 for brass and 1 for stainless steel. Cutting of structures with a higher aspect ratio is possible, but an increasing risk of tool breakage has to be taken into account. By increasing the tool diameter, the aspect ratio of the structures can also be increased, e.g. $100 \mu\text{m}$ wide and $400 \mu\text{m}$ deep grooves in brass. The surface quality appeared to be comparable to that of structures cut with conventional hard metal tools.

The burr at the upper edges of the grooves ranged from a few microns up to more than $20 \mu\text{m}$ (Figures 2, 3 and 4). The burr occurrence depends on the material (low for brass, high for stainless steel), on the cutting edge quality of the tool and on the actual cutting length of the tool.

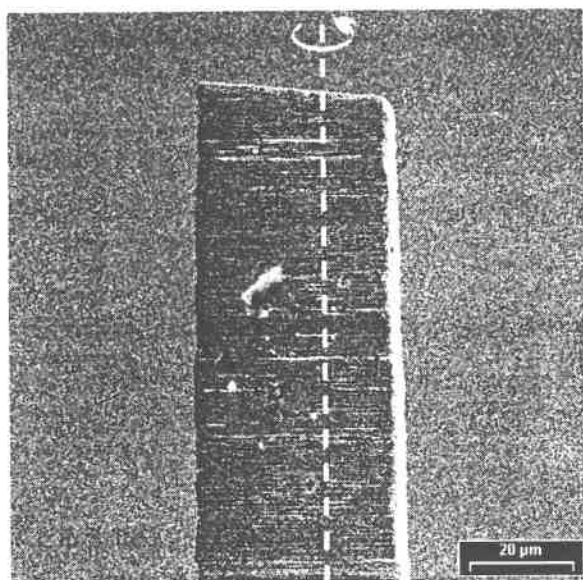


Figure 1: SEM of a ground $50 \mu\text{m}$ hard metal single-edge end mill with cutting edge (left) and clearances (top and right). The dotted line indicates the spindle axis.

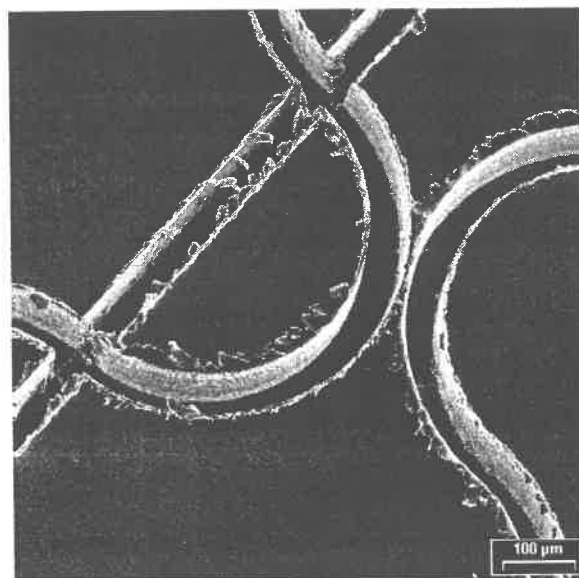


Figure 2: SEM of grooves cut into brass. Minimum groove width is $45 \mu\text{m}$, depth is $115 \mu\text{m}$. Burr occurs at the upper structure edges and at the vertical edges of the groove crossings.

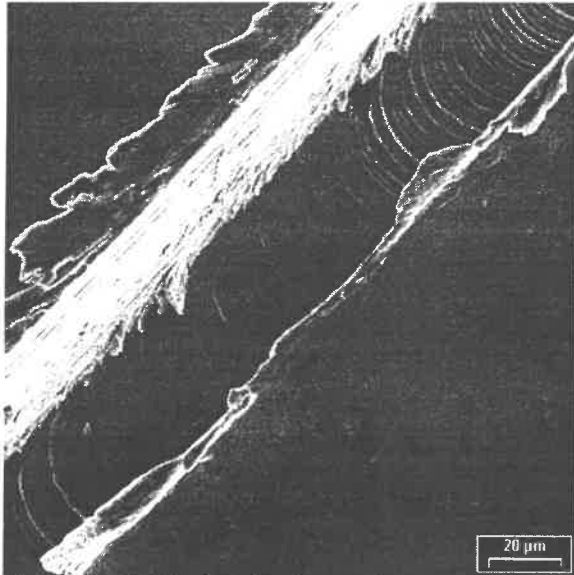


Figure 3: SEM of a structure cut into brass. Burr can be seen at the upper structure edges. The groove bottom and walls are rather smooth, with several brass chips adhered to them.

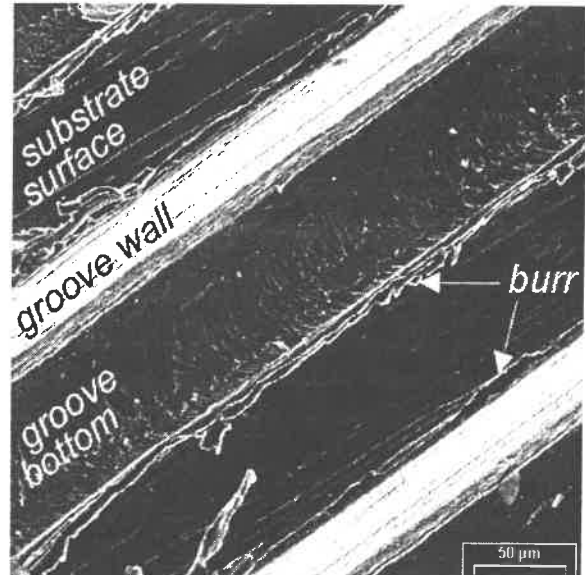


Figure 4: SEM of a structure cut into stainless steel. Burr occurs at the upper groove edges. The lower groove edges are rounded due to the wear of the tool tip.

Especially for the cutting of stainless steel the wear of the cutting edge was evident from the groove profile becoming more and more rounded instead of its initial rectangular shape. The wear resulted in an increasing cutting force and finally in tool failure. The initial radius of curvature of the cutting edge of about 1-2 μm was lost after some 100 millimeters of cutting in stainless steel, and tools with a diameter less than 100 μm usually broke, whereas larger tools could stand a higher cutting length.

For cutting of brass, the tool wear was observed to be negligible.

Cutting lengths until breakage of the tool were determined to be more than 10 m for brass (tool diameter 40 μm and above) and more than 5 m for stainless steel (tool diameter 100 μm and above). Structure quality (groove profile and burr) remained constant for the cutting of brass, but decreased slightly when cutting stainless steel until tool failure. The achieved cutting lengths are sufficient for the realization of microstructured areas of more than 100 mm^2 , depending on the structure depth (Figure 5).

Of course, structures other than the linear test grooves can be cut by using the CNC equipment of the milling machines. An example for the structuring of a 50 μm thick brass foil is given in Figure 6.

Discussion and Outlook

The results show that mechanical cutting of brass and stainless steel with hard metal micro end mills is possible. For microstructure manufacture the structural limit was pushed to groove widths of less than 50 μm with an aspect ratio of 3 (for brass). Although cutting with micro hard metal tools will not replace diamond cutting due to the better surface quality of the diamond-cut structures and the higher cutting length of the diamond tools, hard metal tools can be used for the realization of structural details in small areas. But even molds made of steel – as often requested instead of the less robust usual microstructured molds made of brass – seem to be realizable.

The most recent application of structures cut with 50 μm diameter tools is a microstructured optical mask (Figure 6), used for the further development of an optical particle analysis system [10].

Further tests will be performed to study other hard metals for tools, to cut other materials, such as copper, aluminum and various types of stainless steel and to optimize the cutting length of the tools. A high-frequency spindle will be used for possible structure quality improvement and reduction of the structuring time. Structures cut in brass and stainless steel will be tested in injection molding. First experiments have been made to remove the burr from brass structures by an additional diamond milling step, and promising results have been obtained for burr removal by electrochemical polishing of steel.

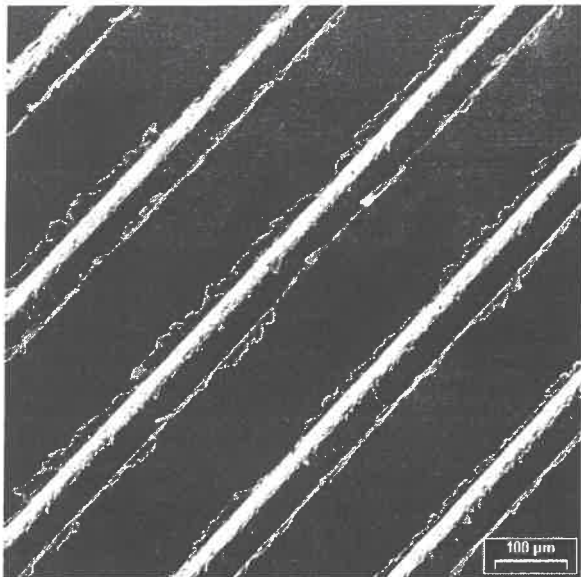


Figure 5: Structured brass surface. The SEM shows part of a 250 mm² area with test structures cut with a single 50 μm diameter tool. Cutting depth is about 100 μm.

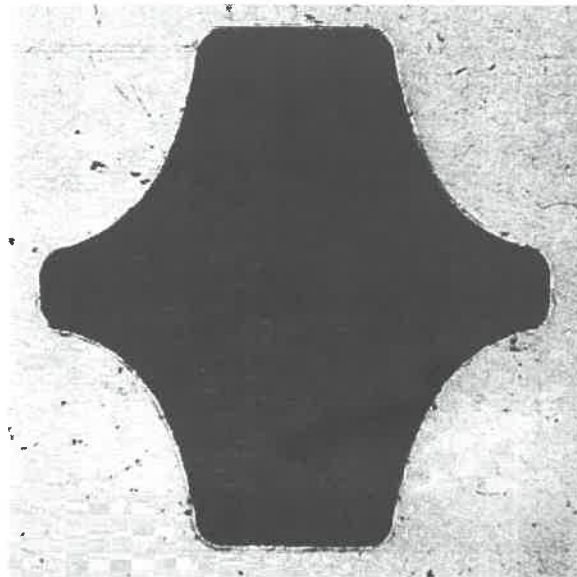


Figure 6: Light microscopy of a hole with hyperbolically shaped borders cut with a 50 μm diameter hard metal tool in a 50 μm thick brass foil.

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Examination of Surface Location Error Due to Phasing of Cutter Vibrations

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Introduction

In computer numerically-controlled (CNC) machining, accurate positioning of the tool with respect to the workpiece for material removal is required. The machine tool's performance can be evaluated by the machine's contouring accuracy, cutting performance and/or repeatability [1]. Sources which cause errors in these categories are geometric and thermal errors, typically termed quasi-static errors, imperfect spindle motions, friction in the drives, controller errors and cutting force, or dynamic, errors. In order to reduce the errors in the machining process, the techniques of avoidance and/or error compensation may be used [2].

The purpose of this research is to investigate the relative importance of spindle speed, system dynamics and cutting conditions on the accuracy of surface location in CNC finish machining operations. The cutter deflections, which produce the surface location errors investigated here, are not easily adaptable to real time compensation. However, as will be shown, proper selection of spindle speed in the NC program may increase the surface location accuracy and act as a type of "NC avoidance".

Forced Vibrations and Surface Location Error

The relationship between the spindle speed, the most flexible natural frequencies of the machine/cutting tool system and the final part dimensions is not a simple one. It has been detailed in [3], [4] and [5] and will be briefly outlined here. As shown in Figure 1, cutter deflections during end milling will only affect the final surface when a tooth is in position 1.

In up milling (Figure 1a), the error of location is defined as the displacement of the cutter relative to the commanded depth of cut when a tooth enters the cut. In down milling, the error is defined as the cutter displacement relative to the desired surface when a tooth disengages from the cut [6]. At other angular orientations of the tool, any imprint of the cutter deflection on the material will be removed by succeeding teeth.

As the cutter impacts the workpiece surface, a situation of forced vibration arises in which the dominant forcing frequency is equal to the tooth passing frequency of the cutter, f_{tooth} (see Equation 1). For simple systems under forced vibration, the deflection varies at the same frequency as the force, but lags it in time. The cutting forces are strongly harmonic and cause deflections of the cutter. The amplitude and phase of these deflections depends not only on the system flexibility and cutting conditions, but also on the system natural frequency (i.e. when the forcing frequency is equal to the natural frequency, resonance is observed and large deflections can occur). The crucial factors, therefore, are the spindle speed (proportional to the tooth passing frequency) and the frequency response of the system. As the tooth passing frequency is varied, the magnitude and phase of the cutter vibration change. This, in turn, varies the final surface location.

$$f_{\text{tooth}} = nm \text{ (Hz)}, \quad \text{where } n = \text{spindle speed in rev/s} \quad (1) \\ m = \text{number of teeth on cutting tool}$$

Over a range of spindle speeds that span an integer increase in the number of waves of vibration per tooth (e.g. $f_{\text{tooth}} = f_n$ to $2f_{\text{tooth}} = f_n$), a periodic change in the surface location will occur. This change in surface location is termed overcut when the part is smaller than commanded (i.e. more material removed than specified) and undercut when larger than commanded.

Consider the case of down milling with a circular tool path assumed. The normal and tangential cutting force components may be projected into the x and y coordinate directions using the cutting angle, ϕ , according to Equation 2. The tangential cutting force, F_T , is taken to be proportional to the material cutting stiffness, K_s , the feed per tooth, f_t , the chip width, b , and the cutting angle. The normal (or radial) force, F_N , is assumed proportional to the tangential force. See Equation 3.

$$F_x = \sum_{i=1}^m F_{T,i} \cos \phi_i + F_{N,i} \sin \phi_i, \text{ where } m = \# \text{ teeth in cut at cutter angle } \phi \quad (2)$$

$$F_y = \sum_{i=1}^m F_{T,i} \sin \phi_i - F_{N,i} \cos \phi_i$$

$$\begin{aligned} F_T &= K_s b f_t \sin \phi \\ F_N &= F_T \cos \beta, \text{ where } \beta = 72^\circ \text{ and } \cos \beta = 0.3 \end{aligned} \quad (3)$$

Substitution of Equation 3 into Equation 2 yields x and y cutting forces, each with three distinct terms: DC, sine and cosine [3]. See Equation 4.

$$F_x = K_s b f_t \sum_{i=1}^m \sin \phi_i \cos \phi_i + 0.3 \sin^2 \phi_i = \frac{K_s b f_t}{2} \sum_{i=1}^m \sin 2\phi_i + 0.3 - 0.3 \cos 2\phi_i \quad (4)$$

$$F_y = K_s b f_t \sum_{i=1}^m \sin^2 \phi_i - 0.3 \sin \phi_i \cos \phi_i = \frac{K_s b f_t}{2} \sum_{i=1}^m 1 - \cos 2\phi_i - 0.3 \sin 2\phi_i$$

For small radial immersions, only a single tooth is in the cut at any given instant. Therefore, the cutting force is approximately constant during engagement and zero otherwise. This force is similar to an impulse with a short duration and gives strongly harmonic frequency content [6]. The frequency spectrum of the cutting force therefore includes both the fundamental tooth passing frequency, ($f_{\text{tooth}} = f_n$), and harmonics which occur at integer multiples of the fundamental frequency.

If one considers only the fundamental frequency and its effect on the surface location error, the three term, y-force component given in Equation 5 may be evaluated and used to find the y-deflection of the cutter (which is imprinted on the final surface at cutting angles which are odd multiples of 180° or π rad for each tooth on the cutter). This y-direction cutter displacement is phase shifted behind the y-force by an angle which is dependent on the ratio, r . Three distinct cases ($r < 1$, $r = 1$ and $r > 1$) can be evaluated to show the variation from overcut to undercut for down milling, however, due to space limitations, the analysis will not be completed here.

Simulation

Although the simple calculations alluded to previously can reveal a picture of the cutter deflections, the determination of the final surface location error is best evaluated by time domain simulation. A simulation for end milling operations was programmed which calculates the tangential and normal forces on the cutter, the resulting cutter deflections in both the x and y-directions and the final error of surface. The simulation also includes regeneration of surface (although it is a secondary effect in forced vibrations [3]) and the non-linearity which arises when the tool jumps out of the cut due to excessive vibrations.

Experimental Results

An experimental part geometry was chosen, the CNC code written and down milling tests completed on a horizontal spindle 3-axis CNC machining center. The workpiece (7075-T6 aluminum) dimensions were measured using a coordinate measuring machine (CMM) in a temperature-controlled environment. The part geometry is shown in Figure 2. This particular geometry doubles the sensitivity to surface location errors because the measurements D1 and D2 each contain two passes which contribute, theoretically, the same surface error. The surface errors, E1 and E2, can then be calculated according to Equation 5. In this equation, a positive error indicates an undercut surface, while a negative error denotes an overcut surface. The commanded values of D1/D2 were 95.174 mm (3.747") for a 12.776 mm (0.503") diameter cutter.

$$E1 = (D1 - 95.174)/2, \quad E2 = (D2 - 95.174)/2 \text{ (mm)} \quad (5)$$

The machining and measurement of 11 total parts corresponding to the spindle speeds/feeds shown in Table 1 were completed. The error of surface location was then calculated according to Equation 5. Additionally, the simulation previously described was executed for the same speeds/feeds to find the predicted error of surface location. A comparison of the experimental and simulated results for both the D1 and D2 surfaces is shown in Figure 3.

It can be seen from Figure 3 that the simulation results match well with the experimental data at speeds of 4313 rpm and above. The predicted overcut to undercut can be seen in both the simulated and D1 surface data with the maximum undercut at 4450 rpm (near the second harmonic of resonance). The D2 surface data shows a probable anomaly at the 4450 rpm spindle speed, but good agreement with the D1 data otherwise. The good experimental agreement between the two surface errors is reassuring since the x and y-direction system dynamics were equal, so the errors should be the same. The discrepancy between the simulated and experimental results at spindle speeds below 4313 rpm (which appears to be a simple offset except at 3766 rpm) could have resulted from several procedural limitations. First, only one part was machined at each spindle speed so the process uncertainty is unknown. Second, only one part was machined per day over an 11 day period. The machine shop temperature variation was significant over this time period, but the measuring environment was kept at a constant temperature of 68° F. Finally, the end mill was removed after each test, so the part could be faced with a separate tool (to limit wear). Therefore, the positioning repeatability of the spindle collet becomes a factor.

Although the simulation and experimental results do not match perfectly, a large variation in the surface location error is recognized over the range of spindle speeds tested in both data sets. A total experimental variation in surface location error of 50 μm (0.002") is shown over the range of spindle speeds for the surfaces under identical cutting conditions using the same NC program. In certain finish machining applications, this is an unacceptable error magnitude. It can also be seen that at spindle speeds of 3770 and 4520 rpm for D1 and 3890 and 4300 rpm for D2 there is no error. Selection of these speeds would produce a part with no error of surface introduced by the cutting operation, even for the flexible tool used in this research.

Conclusions

The effect of spindle speed and the system frequency response function on the surface location error in finish machining has been explored. Cutting tests have been performed over a range of spindle speeds corresponding to a full range of surface location errors from overcut to undercut ($2.65f_{\text{tooth}} = f_n$ to $3.65f_{\text{tooth}} = f_n$) for a given tool and stable cutting conditions. Simulations have also been completed and good agreement found between the simulated and actual data over a limited spindle speed range. Further cutting tests, which include several tools and

various materials and cutting conditions, will be necessary to fully explore this phenomenon. Although not a definitive work, this research demonstrates a significant dependence of the surface location error on spindle speed. Furthermore, it has been observed that a relatively small change in spindle speed can produce a large variation in the final surface location (and part dimensions), especially at speeds near integer multiples of the system natural frequency.

Table 1: Selected Speeds/Feeds

Test Number	Spindle Speed (rpm)	Linear Feed (m/min)
1	4998	2.95
2	4861	2.87
3	4724	2.80
4	4587	2.72
5	4450	2.64
6	4313	2.56
7	4177	2.48
8	4040	2.36
9	3903	2.28
10	3766	2.20
11	3629	2.13

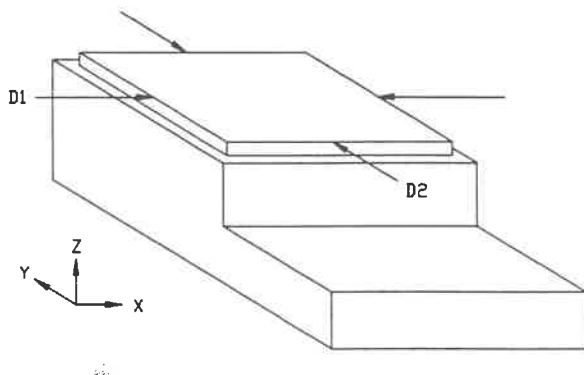


Figure 2: Part Geometry

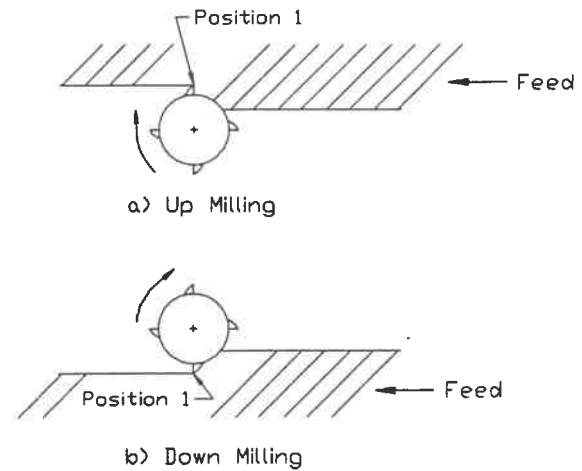


Figure 1: Up/Down Milling

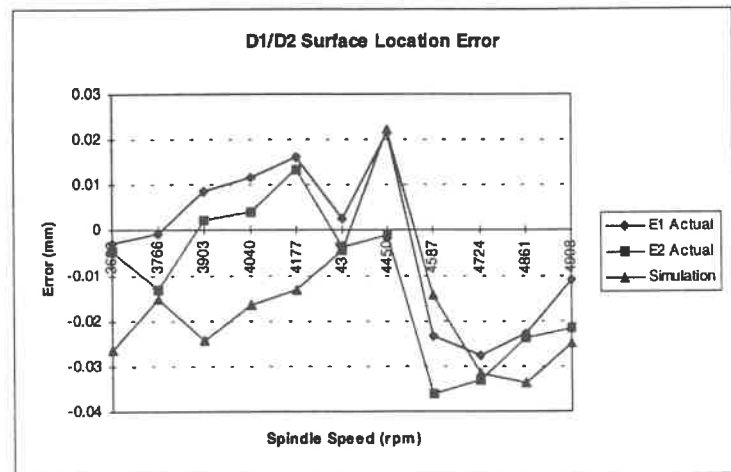


Figure 3: Surface Location Error

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Thermal Aspect of Micro to Nano Scale Metal Cutting

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Background

In cutting process, heat generation in deformed region in workpiece plays an important role as an essential factor affecting chip formation, tool wear and cutting accuracy. Based on the cutting experiments in millimeter to sub-millimeter scale and analysis using a continuum thermal conduction model, the cutting temperature is understood in general to vary in proportion to the square root of undeformed chip thickness [1]. Therefore, the cutting temperature in single point diamond cutting process has been considered to be negligibly low because chip removal in the process is typically performed under very small undeformed chip thickness less than $1\text{ }\mu\text{m}$ or down to nanometric order. On the other hand, gradual wear of cutting edge and oxidization of chip surface, both of which are suggestive of thermochemical process due to temperature rise, can be observed even in nanometric metal cutting [2].

The cutting temperature measured in copper microcutting under the undeformed chip thickness of $1\text{ }\mu\text{m}$ has been reported to be 270 K [3]. However, little knowledge has been obtained about the thermal aspect in micro to nano scale cutting process because of inevitable difficulties in experimental techniques. Furthermore, the microcutting phenomena which take place in a small region can be atomistic in nature rather than continuous as assumed in conventional continuum mechanics.

In this paper, comparative studies on cutting temperature in metal microcutting have been carried out between micrometer scale cutting experiments and sub-micro to nano scale cutting simulations by extended molecular dynamics computer simulations. Effect of cutting temperature on tool wear mechanism is also discussed from an atomistic point of view.

Extended Molecular Dynamics (MD) Simulation

Conventional MD

To analyze the cutting temperature in nano scale, conventional MD simulations are carried out assuming two-dimensional orthogonal cutting of the (111) plane of hypothetical Morse-potential type monocrystalline Cu cut by (111) plane of diamond cutting edge under the undeformed chip thickness of 1 and 5 nm at the cutting speed of 20 m/s [4].

In general, thermal conduction in metal cannot be analyzed correctly in MD simulation, because electron behavior is not taken into account in conventional MD models. As thermal conduction in metals is mainly governed by electron mobility, metals for practical use show considerably higher thermal conductivity and consequently smaller gradient in thermal field than those estimated by MD simulation in terms of lattice vibration. Therefore, the gradient in thermal field is scaled in the simulations in this paper so as to coincide with that calculated by the continuum thermal conduction theory [4][5].

For the analysis of cutting phenomena under steady-state chip removal, the translational calculation area method, by which the cutting distance can be infinitely extended, is used in the simulations [6][7]. In this method, to limit the calculation area in which the behaviors of individual atoms are analyzed by MD method, the boundary and thermostat layers move at cutting speed with the cutting edge. In practical

calculation, the new atoms are successively inserted from the left into the calculation area, in which the cutting edge is fixed, and the atoms left the calculation area are thrown away from the right or top as shown in Fig.1.

Renormarized MD (RMD)

RMD is an extended MD for large scale model, up to micro to millimeter scale. It is characterized by introduction of scale renormalization procedure [8][9]. In this method, an original atomic model composed of huge number of atoms is considered as a collection of atom clusters in which a specified number of atoms are included as shown in Fig.2 (a). By scale renormalization, the size of those clusters is reduced to atomic one and those clusters are handled as if they were particles in conventional MD simulation as shown in Fig.2(b). The equivalent atomic mass and interparticle potentials in the reduced model are determined by a proper conversion from those of original atomic model based on the ratio of scale reduction.

Temperature of individual clusters in RMD can be evaluated from the sum of kinetic energy and internal energy of the clusters. There are two kinds of energy transfer among the clusters due to thermal conduction. One is through particle-particle interaction modified in consideration of thermal conduction by electron [5] in RMD and the other is through diffusion equation. It should be noted, however, that there is the third kind of energy transfer from particle energy to internal energy. This energy transfer is due to 'viscosity'. It is also taken into account by applying viscosity equation (Navier-Stokes equation without pressure term) to the particle system at each time step in RMD. The decreased amount of kinetic energy of particles by this procedure is considered as being changed into internal energy of corresponding particles, and added into the internal energy at each time step.

To analyze the cutting temperature under the undeformed chip thickness of 10 and 100 nm, RMD simulations are carried out assuming two-dimensional orthogonal cutting of Morse-potential type monocrystalline Cu at the cutting speed of 20 m/s.

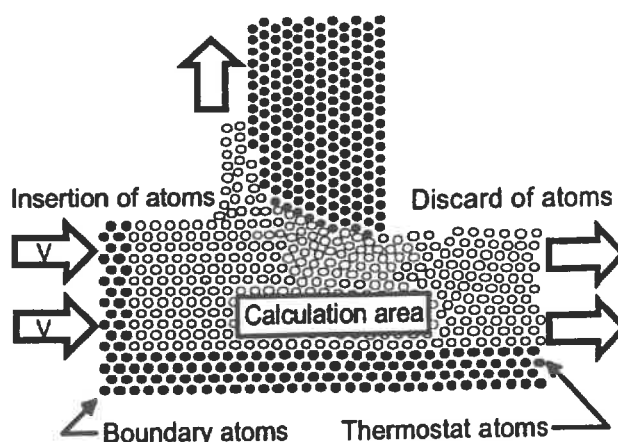


Fig.1 Scheme of translational calculation area method

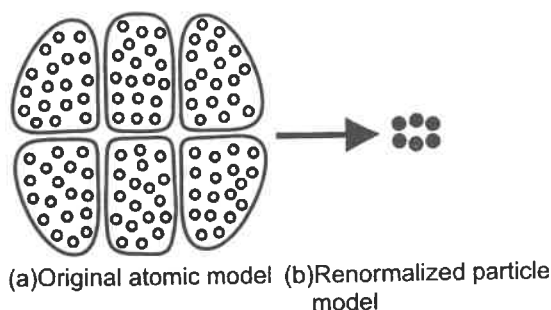


Fig.2 Renormalization process for an atomic model

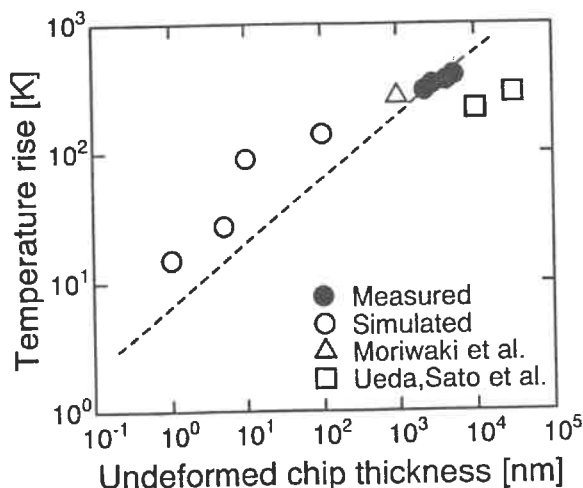


Fig.3 Size effect in cutting temperature in micro-cutting of Cu (cutting speed: 20 m/s, undeformed chip thickness: 5 nm)

Size Effect in Cutting Temperature

Figure 3 shows the relation between the undeformed chip thickness and cutting temperature in cutting of Cu obtained from the extended molecular dynamics simulation in sub-micro to nano scale and micro scale cutting experiments comparing with the measured values reported so far [3][10]. The cutting temperature in micro scale cutting is measured by thermocouple of constantan wire and workpiece. Remarkable size effect can be observed in cutting temperature. The dotted line shows the change in temperature rise extrapolated from the results of the micro scale cutting experiments in proportion to the square root of undeformed chip thickness in accordance with the analysis based on continuum thermal conduction model [1]. The cutting temperature obtained from MD simulations are a little higher than those predicted. In nano scale cutting, temperature rise in shear zone decreases down to 10 to 30 K in average.

Atomistic Cutting Temperature

From the viewpoint of statistical thermodynamics, there is intrinsically a wide scatter in atomistic temperature, which is equivalent to the kinetic energy, of individual atoms in accordance with Maxwell-Boltzmann distribution. Figure 4 shows the distribution of atomistic temperature obtained from MD simulation in the cutting of Cu under the undeformed chip thickness of 5 nm at 20 m/s cutting speed. The temperature of several atoms exceed 3000 K which is corresponding to the kinetic energy of about 0.4 eV. On the other hand, the activation energy is known to be at the order of 0.1 eV for chemical interaction in oxidization or carbonization of practical metals used as the workmaterials for diamond turning. Therefore, the workpiece atoms at this temperature level can supply the activation energy necessary for chemical interaction with the atoms on tool face and/or in atmosphere even if these atoms have no kinetic energy.

When the average temperature of workpiece in contact with the cutting edge is supposed to be 320 K, the probability that the kinetic energy of an atom which is moving toward the tool surface exceed 0.5 eV is calculated to be about 8.6×10^{-10} based on statistical thermodynamics. In cutting of Cu at 20 m/s cutting speed, for example, the number of atoms passing through the effective sectional area of a diamond atom on rake face per unit time is calculated to be about $8.0 \times 10^9 \text{ s}^{-1}$. Therefore, the number of workpiece atom which have a chance to interact with a diamond atom at a specified location on tool face per unit time is estimated at about 6.9 s^{-1} , when the activation energy is supposed to be 0.5 eV. Assuming that each one of these atoms takes away one diamond atom on tool surface, wear rate of

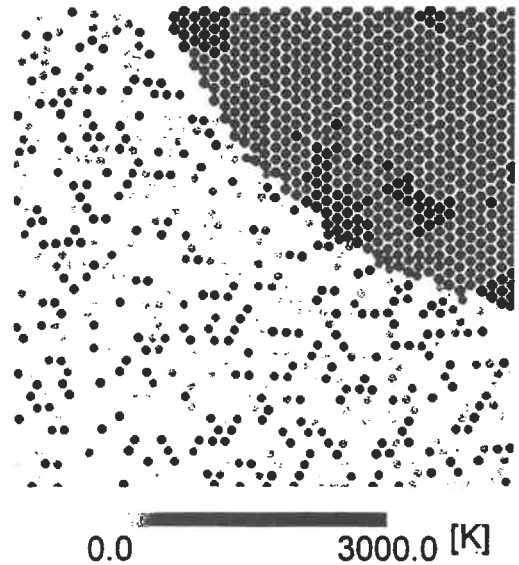


Fig.4 Distribution of atomistic temperature in Cu microcutting (cutting speed: 20 m/s, undeformed chip thickness: 5 nm)

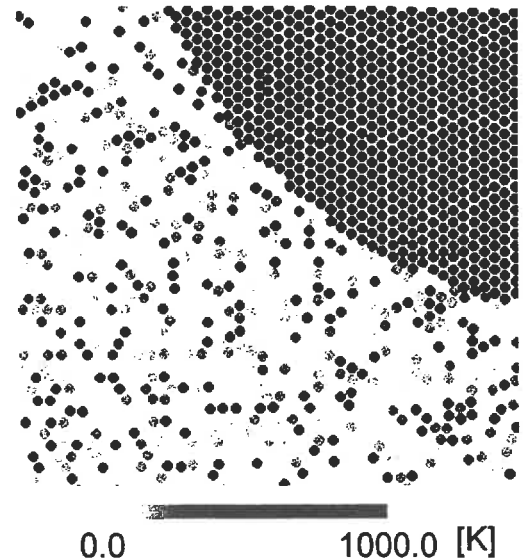


Fig.5 Distribution of atomistic temperature in Cu microcutting at 100 K (cutting speed: 20 m/s, undeformed chip thickness: 5 nm)

diamond tool is roughly estimated at 0.7 nm/s. In other word, this cutting tool can cut the workpiece under the cutting condition for about 30 km cutting distance before the depth of wear reaches to 1 μ m. More accurate quantitative estimation of wear rate can be feasible if detailed information is obtained on interaction process and its activation energy between the atoms of workpiece and tool on their interface by the aid of the analysis based on first principle quantum mechanics. The results also suggest that inevitable difficulty is predicted to avoid gradual thermochemical wear of cutting edge even in nano scale cutting.

From the viewpoint of energy supply for chemical interaction, the most efficient measure to reduce the thermochemical wear of cutting edge is considered to cool the tool-workpiece interface down to cryogenic temperature. Figure 5 shows the distribution of atomistic temperature obtained from MD simulation under the same cutting condition as that in Fig.4 with the temperature control at 100 K in thermostat layers. Only a few atoms the temperature of which exceed 1000 K can be seen in the figure. Under this cutting condition, the probability that an atom have the kinetic energy larger than 0.5 eV is 2.3×10^{-27} . The result shows that negligible tool wear can be expected in the cutting at cryogenic temperature.

Conclusions

Comparative studies on cutting temperature in metal cutting have been carried out between micrometer scale cutting experiments and sub-micro to nano scale cutting simulations by extended molecular dynamics computer simulations. The average cutting temperature rise in shear zone decreases down to 10 to 30 K in nano scale cutting with the remarkable size effect. However, from atomistic point of view, a considerable number of atoms on tool-workpiece interface have the kinetic energy which can supply the activation energy necessary for chemical interaction between workpiece atoms and tool atoms or atmospheric molecules. This results suggest that a thermochemical process is the most probable mechanism of the wear of cutting edge and that the cutting in cryogenic temperature is effective to avoid the wear in nano scale cutting. The quantitative estimation of wear rate can be feasible by the aid of the analysis based on first principle quantum mechanics on interaction process and its activation energy between the atoms of workpiece and tool material on their interface.

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Precision Machining by Hard Turning

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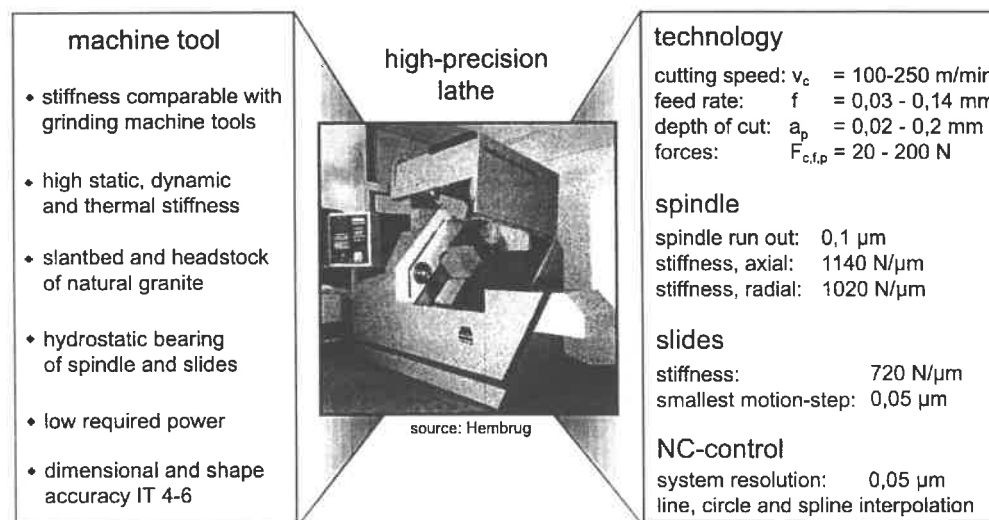
1 INTRODUCTION

In the last years as a main reason the high flexibility of hard turning benefits the substitution of finishing processes with undefined cutting edge. Furthermore the possibility of dry machining means saving considerable costs otherwise caused by buying, treatment and disposal of cutting fluids /TÖN94/. In order to gain acceptance as being an equivalent to present techniques, hard turning must be able to satisfy the increasing quality requirements concerning form and size accuracy, surface finish and surface integrity of the workpiece.

2 MACHINE TOOLS

Up to now in most applications conventional lathes are used for hard turning. These kind of machine tools are often modified in some machine parts to be suited for industrial scale manufacturing. Especially the central components like headstock, tailstock, foundation and slides show higher accuracy, coated linear axis lead to decreased wear of the slideways and linear path-measuring systems realize higher dimension and position accuracy. With such machine tools workpiece qualities in the range of IT 6-7 can be reached under appropriate conditions reliably /KAR97/.

Newly developed high precision lathes offer the possibility to improve the quality of hardturned workpieces significantly (**figure 1**). These machine tools are comparable with grinding machine tools concerning static, dynamical and thermal stiffness as well as accuracy of spindle system and slides /WEC95, LIU96/. Dimensional and shape accuracy in the range of IT 4-6 gets possible by the use of these high-precision lathes /KOC96/. Turning tests on roller bearings with high demands on surface roughness and shape accuracy show the potential of precision hard turning.

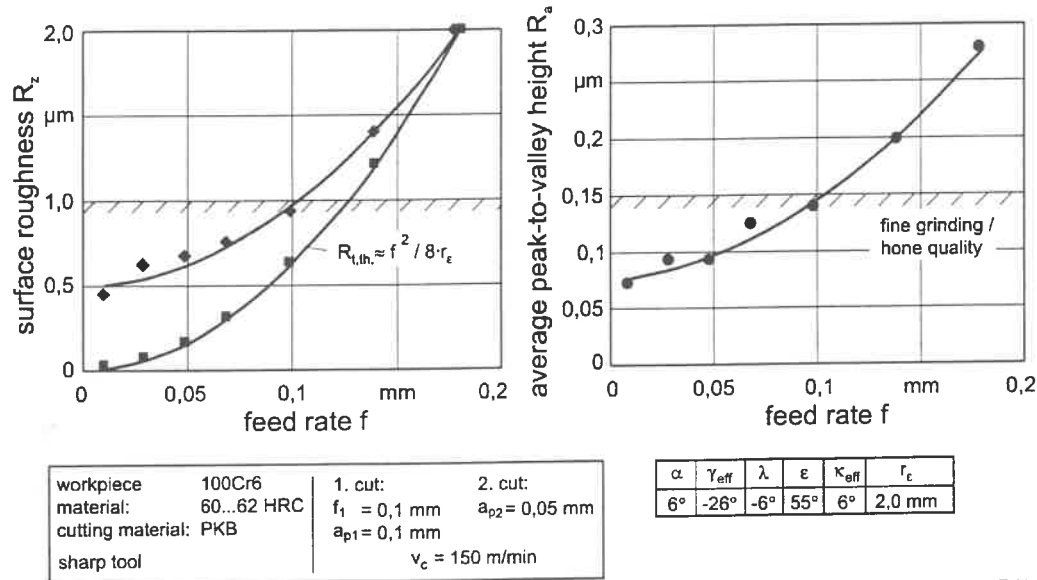


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Figure 1: Characteristics of high-precision lathes suited for hard turning

3 WORKPIECE QUALITY

A main criteria of the workpiece quality especially for finishing processes is the surface roughness. In many applications conventional lathes deliver sufficient surface qualities in the range of $R_z = 2\text{--}4\text{ }\mu\text{m}$. Higher requirements on surface quality can be fulfilled by the use of high-precision lathes. In this case, like **figure 2** shows, surface roughness in the range of $R_z = 0,5\text{--}1\text{ }\mu\text{m}$ can be achieved [TÖN97].



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Figure 2: Surface quality on high-precision lathes

Besides surface roughness, for some precision components like bearings or gear parts, the surface microprofile is of great importance for functional behavior. These precision parts require a very fine and smooth contour. To realize these microprofiles high-precision lathes are necessary, as **figure 3** shows.

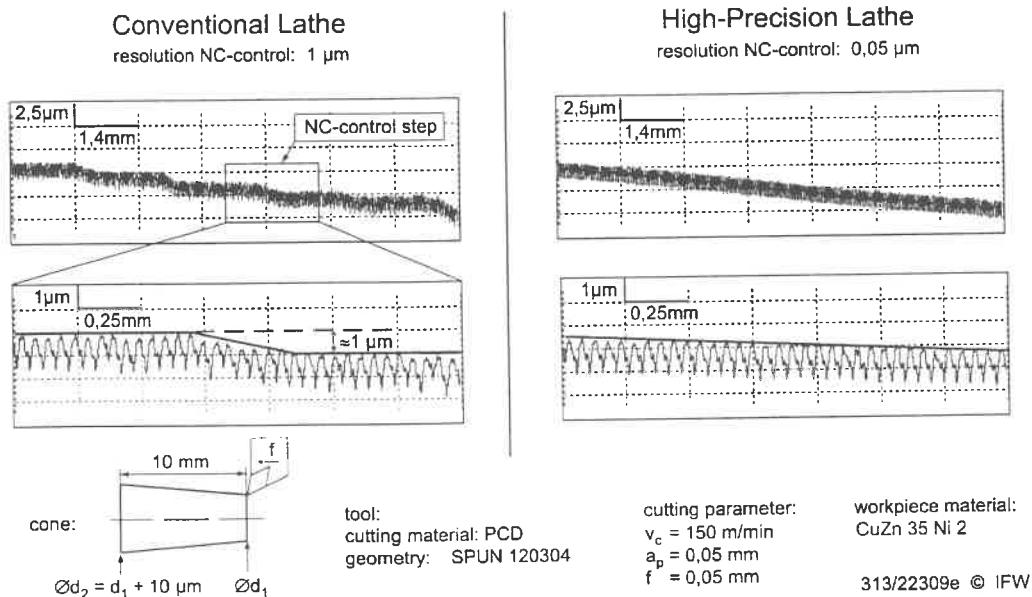


Figure 3: Surface microprofile in cone turning

By use of conventional lathes steps in the microprofile in the range of the nc-control resolution occur. In this case about $1\text{ }\mu\text{m}$, which is a typical value for conventional lathes. By the use of a high-precision lathe with a nc-control resolution about $0,05\text{ }\mu\text{m}$ no steps are recognizable.

The ability of the slides to move in steps in the range of $0,05\text{ }\mu\text{m}$ without stick-slip behavior because of the hydrostatic bearing is the pre-condition to achieve a convex microprofile. Due to this property it is possible to generate microprofiles with high shape accuracy e.g. required by roller bearings to avoid unfavorable edge pressures.

As **figure 4** shows, a smooth profile without recognizable nc-control-steps after machining can be realized. Hence, cylindrical roller bearings permit hard turning as a finishing process.

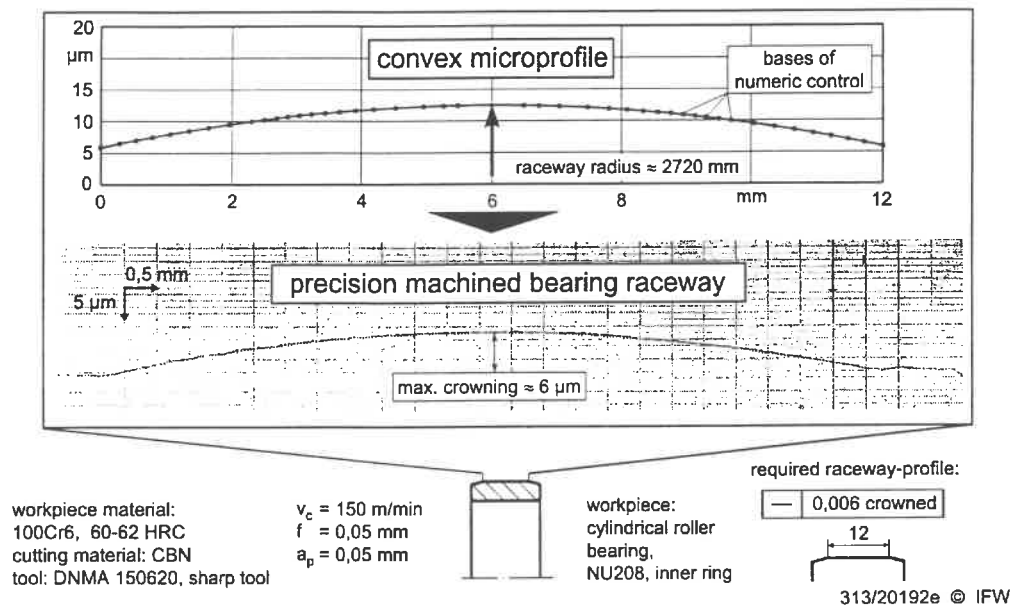


Figure 4: Microprofile of a cylindrical roller bearing raceway

In order to machine this microprofile bases of the theoretical path are implemented in the nc-control. With linear interpolation of these bases the required profile can be realized.

A main bonus of hard turning is avoiding cutting fluids. The possibility of dry machining means saving considerable costs otherwise caused by buying, monitoring, treatment, and disposal of cutting fluids.

On the other hand in case of dry machining form errors gain more importance. Especially in regarding hard turning as a finishing process with lower tolerances to fulfill. Form errors occur due to thermal expansion of workpiece and cutting material and they grow up with increasing cutting length and tool wear /LIE98/. In outer diameter machining the thermal expansion of the workpiece leads in feed direction to a reduction and in inner diameter machining to an increasement of the workpiece diameter. The thermal effect leads to deviation of parallelism of the surface lines.

Newly developed high-precision lathes offers the possibility to compensate these form errors, because of the ability to move in steps in the range of $0.05\text{ }\mu\text{m}$. This is the pre-condition to implement a correction in the nc-control and to receive a smooth profile without recognizable nc-control-steps after machining.

Figure 5 gives a example about compensation of form errors due to thermal expansion, in machining the microprofile of the already mentioned cylindrical roller bearing.

For the compensation a correction reference profile with different gradients, described by the correction factor K , is used. In case of K about $20\text{ }\mu\text{m}$ the parallelism of the surface lines is in the required range of $3\text{ }\mu\text{m}$. Lower or higher K - factors lead to unallowed deviations of parallelism. Thus, by the use of high-

precision lathes it is possible to compensate form errors, above all in case of dry machining, to fulfill the high demands on workpiece quality.

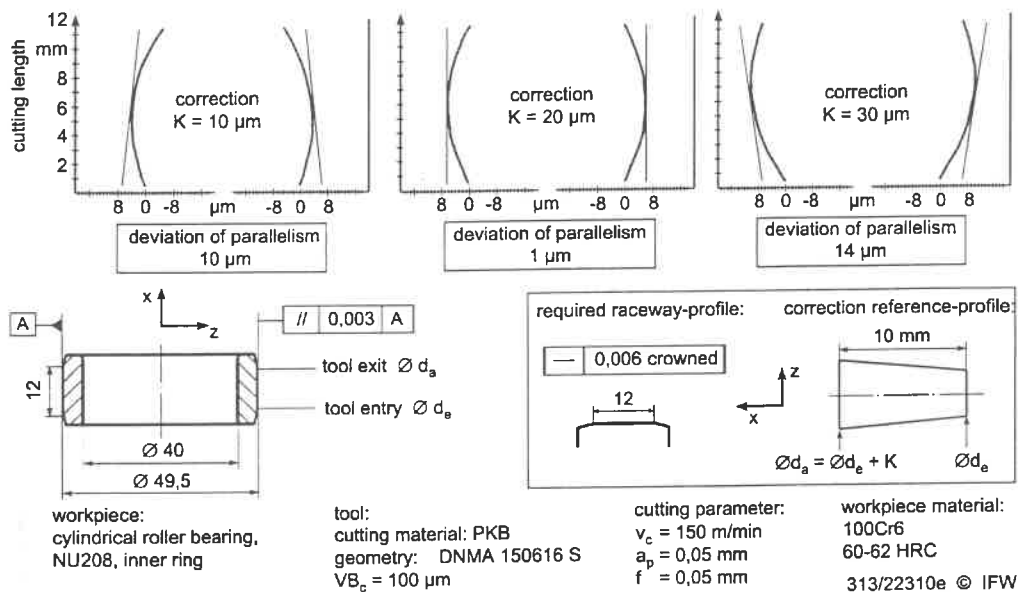


Figure 5: Compensation of form errors due to thermal expansion

4 CONCLUSIONS

Newly developed high precision lathes offer the possibility to improve the quality of hardturned workpieces significantly. Turning tests on roller bearings with high demands on surface roughness and shape accuracy show the potential of precision hard turning. Due to the ability of high-precision lathes to move in steps less than 1 μm especially fine and smooth microprofiles can be realized which are necessary for some precision parts. Furthermore compensation of forms errors as a result of thermal expansion of the workpiece gets possible.

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RE-SHARPENING ENDMILL SIDE-EDGE ON A MACHINING CENTER

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1. Introduction

When dressing the grinding wheel, the edges can be uniform, since the dressing method is performed on a machine tool with the wheel mounted and rotated. On the other hand, the endmill is re-sharpened by a tool grinder, and then mounted on the main spindle of a machining center. Therefore the cutting edge has a run-out based on the spindle rotation and the chucking error.

This run-out of the cutting edge of a endmill has adverse effect on the machined accuracy and surface roughness through a change in the depth of cut. As to the machining accuracy, the problem has been solved by measuring the blade radius of a rotating endmill on the machining center, and by development of a correct-machining process. However, the run-out is a great factor for the roughness of the finished surface. The cutting blades of different radius of gyration makes the surface roughness several times higher than the blades of even radius.

The radius of gyration of the blade is best evened up by forming the endmill when it is installed at the main spindle of a machining center. This research presents the technique with which the radius of gyration of each blade is uniformly formed by re-sharpening the endmill on the machining center. And a personal-computer controlled machining center, which builds in the measurement function and the grinding function, is made for trial purposes to prove the improvement of the finish accuracy by

the on-machine re-sharpening of endmill.

2. Shape measurement and grinding method

The concave method is adopted for this experiment to form the flank of the side blade of the endmill. This method has benefits in that the inclination of the grinding wheel need not be controlled and sharp blades suitable for micro milling can be formed.¹⁾

The endmill blade position can be specified by the distance from the main spindle center, the angle from the rotation origin and the height from z-axis origin. To grind the side blade with relief angle α , rotating the endmill by angle ϕ to make the blade come on the x-axis, the center of the grinding wheel of R_g in radius must be moved to the position (x_1, y_1) shown in the next equation.

$$x_1 = R + R_g \cos \alpha \quad (1)$$

$$y_1 = -R_g \sin \alpha \quad (2)$$

where the main spindle center is assumed to be X and Y axis origin.

In order to uniformly grind the whole area of a spiral blade, the tip of the blade must be kept on the x-axis synchronizing z-direction feed with the rotation of the endmill. z-axis position z_1 is given by the following equation. In the equation, θ represents lead angle of the endmill blade and ϕ_0 is ϕ at z-axis

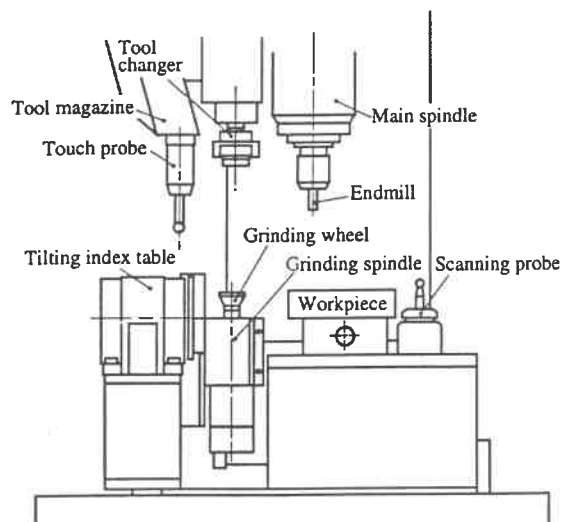


Fig. 1 Measurement and re-sharpening unit on machining center

origin z_0 .

$$z_1 = z_0 + (\phi - \phi_0)R \cot \theta \quad (3)$$

θ , ϕ_0 and R used by the expression should be already-known to re-grind the endmill blade. These values will henceforth be called the shape parameters of the endmill. Because θ of the shape parameters is a peculiar value to the endmill, the design value is adopted to calculate z_1 . R and ϕ_0 are measured with a measurement unit. And the radius of the grinding wheel R_g is measured with a touch probe as described later.

3. On-machine tool grinding system

The on-machine re-sharpening system consists of a measurement unit which measures the shape parameters of the endmill, a grinding unit which re-sharpens the endmill, and a control unit which calculates grinding sequence using the shape parameters and controls the whole system. Figure 1 shows the arrangement of the measurement unit and the grinding unit on the machining center table. Table 1 shows principle specification of the machining center with on-machine re-sharpening system.

For the measurement unit, a three dimensional scanning probe with the resolution of $1\mu m$ is used. The probe is set on the machining center table and

Table 1 System configuration

Machining center	
Capacity:	X*Y*Z: 200*150*200
Taper hole:	7/24, No.20
Control unit:	board type NC controller
CPU	80486 133MHz, OS MS-DOS 6.2
Feed resolution:	10mm/32768pulses
Spindle resolution:	360deg/16384pulses
Re-sharpening unit	
Grinding wheel	
$\phi 50mm$ Cup type	
Resin bond diamond #800	
Revolution speed(max):13000rpm	
Dressing wheel (attached on main spindle)	
$\phi 100mm$ GC wheel #100	
Measurement unit	
Tool measurement system	
Renishaw scanning probe	
Resolution: $1\mu m$	
Grinding wheel measurement system	
Touch probe (attached on main spindle)	
Repeatability: $1\mu m$	

moved to the measurement position according to a machining center feed command. Indexing the main spindle every 0.17° , the profile of the endmill is measured by the scanning probe.

It is necessary to measure the grinding wheel radius R_g after the wheel is dressed. The wheel radius can be indirectly measured by measuring the endmill diameter after grinding. But in this experiment, a touch probe is installed at the main spindle head.

The grinding unit is set up on the machining center table with the grinding wheel spindle parallel to the main spindle of the machining center. The grinding wheel is driven with a high frequency motor to achieve a necessary peripheral speed with a grinding wheel of small diameter. In time of dressing, the dressing wheel is installed at the main spindle of the machining center.

A personal-computer base CNC controller is used as a control unit, which simultaneously controls X, Y and Z-axis and rotating angle of the main spindle. In addition to this, the control unit obtains data from the measurement unit and synchronizes

Table 2 Experimental conditions for re-sharpening

Target tool	High speed steel solid endmill $\phi 10\text{mm}$, $l 50\text{mm}$ Two blades, Lead angle 30° .
Re-sharpening conditions	Grinding speed: 1257m/min Feed speed: 180mm/min Depth of cut: $20\mu\text{m}$ (maximum) Spark-out: none

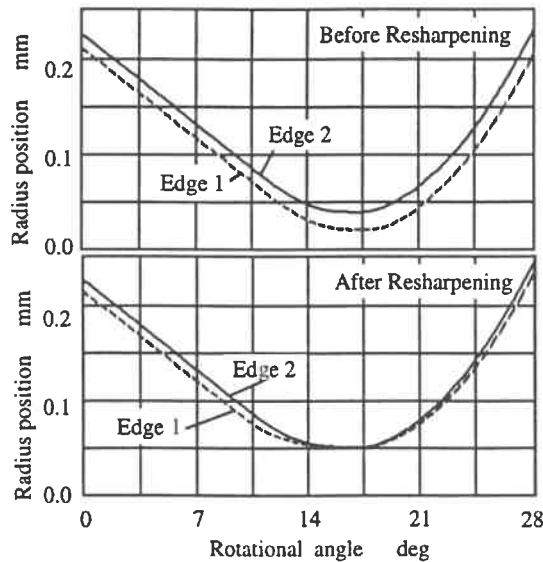


Fig. 2 Edge profiles before and after re-sharpening

the measurement and grinding behavior.

4. Experiment of on-machine re-sharpening of endmill

The experiment to determine the shape parameter and re-sharpening of a endmill is carried out using the above-mentioned system. The target endmill is high-speed steel 2 blade endmill of 10mm in diameter and 30° in lead angle. The measurement condition and the grinding condition are set as shown in Table 2.²⁾

For a grinding wheel, a resin-bonded diamond wheel of #800 in grain size and 50mm in diameter is used. The whole flank area of the endmill blade is ground by the grinding wheel rotating 8000rpm and z-direction feeding with 180mm/min . The grinding wheel is dressed by a GC grain dressing wheel of 100mm in diameter attached to the main spindle of the machining center.

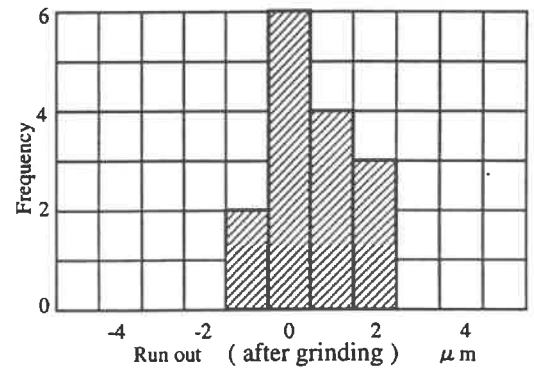
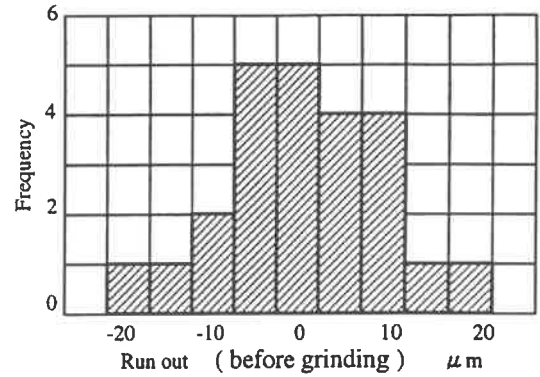


Fig. 3 Distribution of run out

As the typical run-out is $20\mu\text{m}$ or less, the depth of cut is set to finish the re-sharpening with a single pass. Moreover, the spark-out is omitted because the tool shape is not changed by the spark-out. As a result, the net grinding time is 16s for one blade of 50mm in length.

5. Experiment result and consideration

The shape of the blade before and after re-sharpening is measured. figure 2 shows the plot of the output of scanning probe. The measurement position is 1mm from end of the endmill. The flank side of the tool is observed to be uniformly ground and the radius of both blades are corresponding after re-sharpening. It takes 10 seconds to measure the blade form within the range shown in figure 2.

The run-out error is measured by repeating the operation of removing the endmill from the collet chuck and re-installing it back again. As a result, the distribution of run-out of the blade shown in figure 3 is obtained. This figure shows the distribution of difference between the radius of gyration R_1

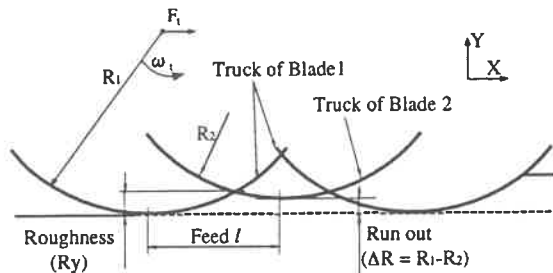


Fig. 4 Surface generation by endmill with run out

of blade 1 and the radius of gyration R_2 of blade 2 as a histogram.

Run-out of the blade after the tool is installed is distributed $-20\mu\text{m} \sim 20\mu\text{m}$. On the other hand, the run-out of a re-sharpened blade became $-1\mu\text{m} \sim 2\mu\text{m}$. Hence it can be seen that on-machine re-sharpening improves the run-out error to 1/10 compared to that observed at initial installation.

The surface roughness generated by machining using an endmill with a run-out is investigated. Figure 4 shows geometrical generation of surface roughness. When the endmill is fed at the rate of $f\text{mm/blade}$, the tracks of two blades, with radii R_1 and R_2 respectively ($R_1 > R_2$), are shown in expression (4) and (5) using parameters α_i .

$$x_1 = l\alpha_1/\pi \pm R_1 \sin \alpha_1, \quad y_1 = R_1 - R_1 \cos \alpha_1 \quad (4)$$

$$x_2 = l\alpha_2/\pi \mp R_2 \sin \alpha_2, \quad y_2 = R_1 + R_2 \cos \alpha_2 \quad (5)$$

where, double sign takes an upper sign when upward cutting and a lower sign when cutting downward.

Geometrical maximum surface roughness is obtained at the Y coordinate value of the intersection of the curves of the expression (4) and (5). If the run-out exceeds the Y coordinate value when the trace of blade R_1 intersects itself, the blade of R_2 does not take part in cutting and it becomes equal to cutting with one blade endmill.

Surface roughness obtained with endmills of different run-out is measured and compared with the theoretical maximum roughness calculated by the above equation. The results given in figure 5 shows that the measured values of surface roughness compare well to the theoretical values. Hence it is shown that surface roughness is improved by reducing run-out of the blade.

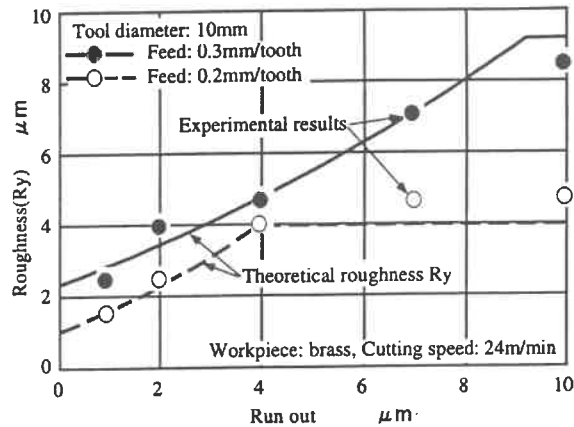


Fig. 5 Relationship between run out and surface roughness (downward cutting)

6. Conclusions

This study aims to develop an on-machine re-sharpening system that removes the overall run-out of endmill blades and enables precise processing. For trial purposes, a machining center that has a measurement and re-sharpening function for a endmill installed in a main spindle is built.

The output signal from the measurement unit is used together with the positioning mechanism of the machining center in order to measure the shape parameter of the endmill. Re-sharpening path for the endmill outer blade is calculated based on the measured shape parameters, and the blade is re-sharpened by the concave method with a grinding unit on the machining center.

As a result, the difference of the radius of gyration of two blades is able to be improved to less than $2\mu\text{m}$. This value equals 1/10 of the run-out of the unprocessed endmill blade just after being installed at the machining center main spindle. It is demonstrated that excellent surface roughness can be obtained by reducing run-out of the blade of the endmill.

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Brittle-Ductile Transition Behavior in Diamond Turning of Single Crystal Silicon Using Straight-Nosed Tools

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1. Introduction

Single crystal silicon is not only a dominant substrate material for micro-electronic and micro-mechanical components, but is also an important infrared optical material. The need for ductile regime machining of silicon to obtain a mirror surface without subsequent polishing has grown remarkably. The essential mechanism underlying ductile regime machining is the brittle-ductile transition. However, various important aspects of the brittle-ductile transition process have not yet been clarified.

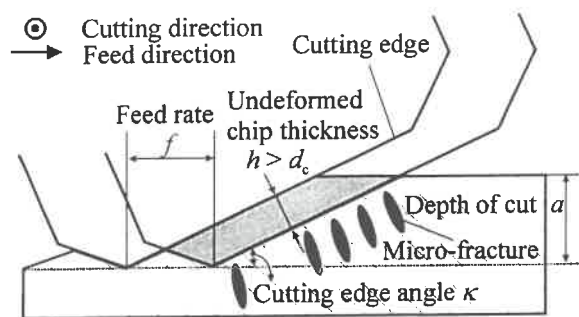
Most of the available literature on silicon diamond turning is concerned with the round-nosed diamond tool¹⁾⁻⁴⁾. On the occasions of round-nosed tool turning, the undeformed chip thickness varies along the cutting edge so that a transition point exists on the cutting shoulder^{1) 2)}. Studies on brittle-ductile transition have been performed by examining the position of the transition point on the shoulder. However, the machining model for the round-nosed tool becomes complicated due to crack history effects^{1) 2)}. Additionally, it is difficult to grasp the respective machining characteristics of the brittle regime and the ductile regime due to the coexistence and interference of those two regimes. In the present study, ductile regime turning using a straight-nosed diamond tool is proposed. The material removal behavior, cutting force and the energy dissipation in brittle-ductile transition of silicon are examined.

2. Geometrical machining model

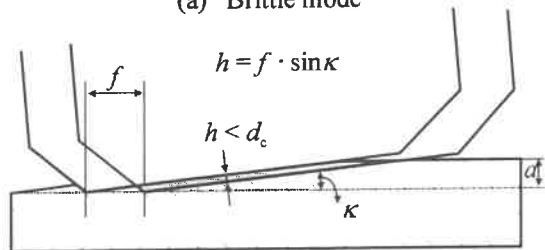
The geometrical machining model of the straight-nosed diamond tool is shown in Fig.1. In this model, the cutting mode becomes almost uniform along the entire width of the cutting edge. By precisely controlling the cutting edge angle κ , an extremely small undeformed chip thickness h can be obtained even at a relatively large tool feed rate f . This enables a kind of large feed rate ductile regime machining, which decreases tool wear and machining time, and subsequently machining cost. Moreover, straight-nosed tool turning provides an effective method for studying brittle-ductile transition due to the simplification of the machining model. The critical depth of cut d_c can be obtained directly by determining the critical feed rate on the machined surface. Furthermore, consistency in chip thickness contributes to the characterization of chip morphology and the evaluation of specific machining energy dissipation.

3. Experimental procedure

The present experiment is conducted on a TOYODA ultra-precision diamond turning machine having a hydrostatic bearing spindle. A precision rotation tool post enables the fine adjustment of the cutting edge angle in B axis. A piezoelectric dynamometric sensor (Kistler 9251A) is installed into the tool holder to measure the micro-cutting force. A straight-nosed diamond tool having a 1.2 mm main cutting edge, a -20° rake angle and a 6° relief angle is used for cutting. Before cutting, the tool is examined using a scanning electron microscope. The cutting edge sectional profile is measured using SEM-based 3-dimensional analysis equipment which has two detectors, respectively for detecting secondary electrons and reflection electrons. The edge radius is estimated to be 50 nm, as shown in Fig.2. Silicon (111) wafer is used as the specimen. Undeformed chip thickness is varied from 0 to 1 μm by adjusting the cutting edge angle and the tool feed rate. Dry cutting is performed while varying the cutting speed in the range of 1.57~3.14 m/s. The machined surface is examined using a Nomarski differential interference contrast microscope, and the chips are observed using the scanning electron microscope.



(a) Brittle mode



(b) Ductile mode

Fig.1 Geometrical machining model

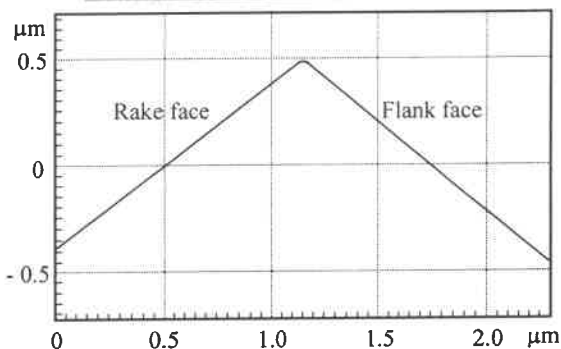
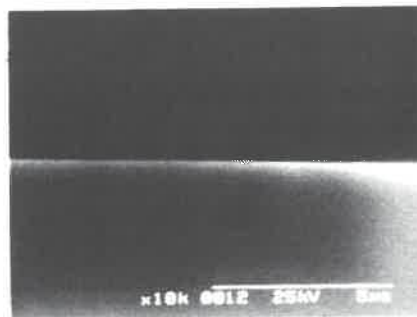
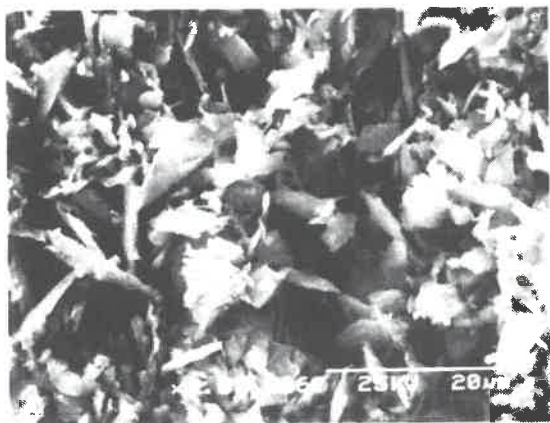


Fig.2 SEM image and sectional profile of cutting edge

4. Results and Discussion

4.1 Chip morphology and surface texture

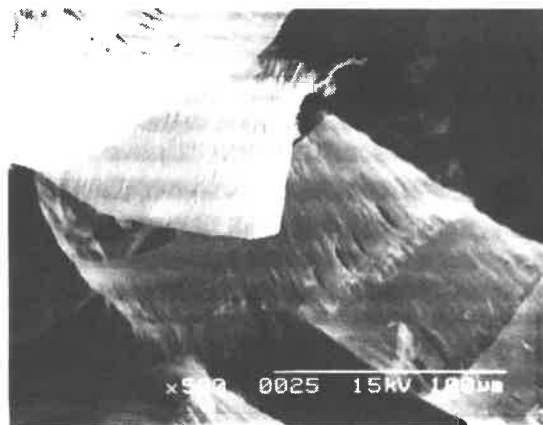
Chip morphology varies significantly with respect to undeformed chip thickness h . Figure 3 shows photographs of typical chips. Fig.3(a) shows the chip of $h = 868$ nm, which is composed of irregular pieces and grains of micron order size. Fig.3(b) shows the chip of $h = 163$ nm, which appears to be formed as stacked lines. Fig.3(c) shows the chip of $h = 58$ nm, which is in the form of a continuous ribbon. Fig.3(d) shows the chip of $h = 12$ nm, which exhibits a sponge-like loose microscopic structure. The machined surface corresponding to chip (a) is severely damaged. The surface corresponding to (b) is slightly pitted and has shallow striations perpendicular to the cutting direction. The surface corresponding to (c) is smooth and has no visible damage. The above mentioned characteristics of the chips and their respective surfaces are found to occur sequentially as the undeformed chip thickness is decreased, although differences in chip appearance are observed even for the same h , which is due to variation in crystal orientation. In addition, the undeformed chip thickness at which the damage-free surface begins to form in all orientations is found to be approximately 90 nm, which is considered to be the critical depth of cut for this wafer.



(a) $h = 868$ nm



(b) $h = 163$ nm



(c) $h = 58\text{nm}$



(d) $h = 12\text{nm}$

Fig.3 Typical chip morphology

4.2 Cutting force and specific energy

Figure 4 shows the variation in principal cutting force F_c and thrust cutting force F_t with respect to undeformed chip thickness h . Both F_c and F_t increase rapidly with respect to h when h is small. After reaching maximums at 90 nm, F_c and F_t decrease simultaneously, eventually reaching values that do not vary noticeably when h is further increased. Fig.5 shows the variation in specific machining energy u with respect to the undeformed chip thickness h . It is clear that u increases rapidly as h decreases, especially when $h < 200$ nm. The h - u curve presents a relatively larger slope around $h = 100$ nm, where the curve intersects an exponential fitting curve that characterizes the size effect of specific energy in metal cutting⁵⁾.

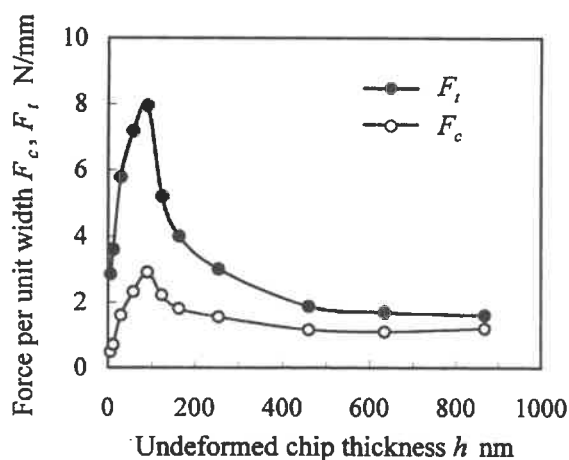


Fig.4 Cutting force versus h

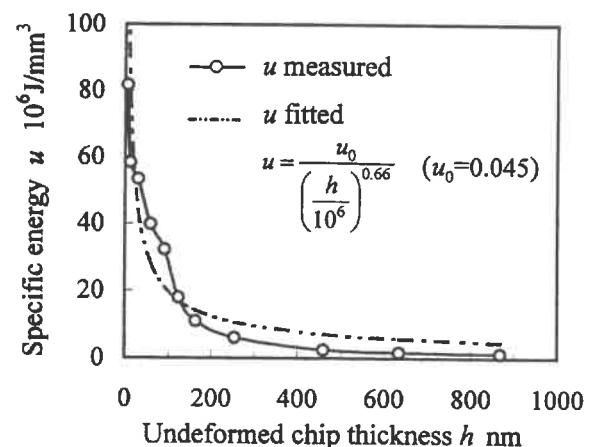


Fig.5 Specific energy versus h

4.3 Discussion

Variations in chip morphology and surface texture support the classification of cutting modes into three categories for the undeformed chip thickness: brittle mode, intermediate mode and ductile mode. The respective material removal characteristics are assumed to be as described below. When the undeformed chip thickness is large, micro-fracture occurs. The creation and propagation of cracks leads to crushed irregular chips and damaged surfaces, which is similar to the typical material removal pattern in cutting ceramics⁶⁾. In the intermediate mode, cracks do occur, but both the quantity and the randomness of the cracks are suppressed. Here cracks tend to initiate and propagate near the shear plane, and periodical shear deformation occurs while generating the segmented chip units. In addition, this shear deformation is liable to be synchronized in the direction of the cutting edge because the undeformed chip thickness is uniform along the edge. As a result,

the linear chip is formed and traces of the shear deformation are left as striations on the machined surface. Subsequently, when the undeformed chip thickness is decreased sufficiently, plastic deformation dominantly occurs rather than crack generation, which is considered to be ductile regime machining.

An analysis of the specific energy reveals another important aspect of brittle-ductile transition, i.e., in order to remove a certain amount of material, a larger quantity of energy is expended in the ductile regime than in the brittle regime. In the brittle regime, the machining energy is mainly converted as surface energy due to crack surface propagation whereas little plastic deformation occurs. However, in the ductile mode, the material is intensively deformed in not only the removed chip but also the subsurface layer^{7) 8)}, which leads to a high specific energy level. It is reasonable to assume that the specific energy is related to the proportion of deformed material volume present in the total removed material volume.

In addition, when the undeformed chip thickness is decreased to an extremely small value, the chip formation behavior differs from that of the traditional cutting phenomenon. This regime tends to generate a loose chip structure rather than the solid ribbon chip. This is considered to be a result of the excessive negative effective rake angle induced by the tool edge radius, which generates a large compressive stress field and a large specific energy level in the cutting vicinity. The chip formation process may involve the multiple effects of stress release, high pressure phase transformation⁹⁾ and tool-work friction etc.. The machining mechanism in this regime is under further investigation.

5. Summary

Diamond turning using a straight-nosed tool provides an effective means to study the material removal behavior and energy dissipation in brittle-ductile transition of brittle materials such as single crystal silicon. Microscopic observations of chip morphology and surface texture are performed and cutting force is measured. The brittle-ductile transition process is found to involve an intermediate cutting regime, in which periodical shear deformation occurs. Continuous ribbon chips are formed in the ductile regime. When the undeformed chip thickness is decreased to an extremely small value, the chip tends to have a loose micro-structure. A remarkable increase in the specific energy is manifested in the brittle-ductile transition beyond the conventional size effect.

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ULTRAPRECISION MACHINING OF COPPER-BERYLLIUM ALLOYS

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INTRODUCTION

Investigation on ultraprecision machining has increased since 1960's. Although ultraprecision machining of many nonferrous alloys such as aluminum and polymers was investigated, there were only few papers devoted to commercial copper alloys. Most published studies used oxygen-free copper alloys that might include silver and boron. Copper beryllium (Cu-Be) alloys were used extensively in electronics industry for its high conductivity, modulus, and strength; they were also popular in optical industry for other laser or infrared applications. Information on micromachinability of Cu-Be, therefore, was increasingly demanding. The objective of this study was to investigate the factors that affect the nanometric surface quality of Cu-Be alloys due to ultraprecision machining.

EXPERIMENT

The nomenclature used here followed the recommendation of the SAE Unified Numbering System. The materials utilized in this study were C17300 (Cu, 1.84 wt% Be, 0.23 Co, 0.02 Ni, 0.04 Fe, 0.1 Si, 0.03 Al, 0.24 Pb), and the non-leaded alloy C17200. Large rods were solutionized at 820°C, water quenched, then cold-drawn ~35% to full-hard (H temper) to produce final rods of 9.525 mm diameter. Some rods were peak-aged at 316°C for 2 hrs (HT temper) after being cold drawn. Both continuous and interrupted facing of short Cu-Be coupons were performed on a commercial ultraprecision machine that had 2.5 nm positioning accuracy. Depth of cut was varied from 0.2-1.0 μm . All feedrates were normalized as the percentages of tool nose radii. Both single crystalline diamond tool (SCD, with (100)-rake face and 20-70 nm edge sharpness), and polycrystalline diamond tools (PCD, with 0.5 μm grains, 500-750 nm edge sharpness) were used for the facing operations. The cutting forces were monitored with a dynamometry system that had 10 mN resolution. A sample surface was analyzed with a Digital Instrument scanning probe microscope (SPM), a Form TalySurf profilometer (FT), a laser interferometer and a scanning-electron microscope (SEM).

RESULTS AND DISCUSSION

Effect of microstructure

Cobalt and nickel were added in the material to promote a fine-grained structure in Cu-Be alloys. These elements, however, combined with beryllium during solidification to form beryllide inclusions. Although these large beryllides were broken up due to a high shear stress during cold drawing, they were not completely dissolved during the solutionizing process. The microstructure of Cu-Be included copper grains, beryllides, lead particles, and additional γ precipitates in an aged sample. The presence of micron-size beryllides degraded the micromachinability of Cu-Be. When collided with a single-crystalline diamond tool, the beryllides were broken and either imbedded in the chip (Fig. 1) or smearing along with the tool and plowing on the machined surface. It was postulated that such collision also chipped off a sharp and brittle diamond cutting edge (20-70nm). Tool chipping and smearing/plowing of broken beryllides worsened the surface finish of a machined sample. The presence of the

large inclusions, nearly $1\text{ }\mu\text{m}$ as seen in the distribution in Fig. 2, perhaps was the reason for the random "excessive tool wear" observed when micromachining different batches of Cu-Be as reported in literature [Arnold et al, 1975].

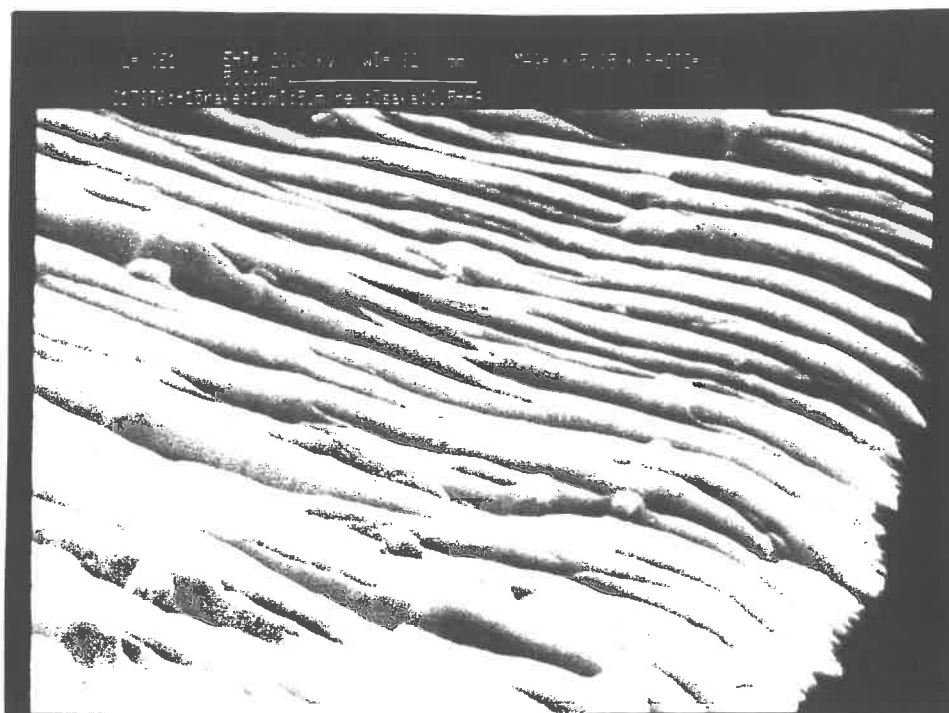
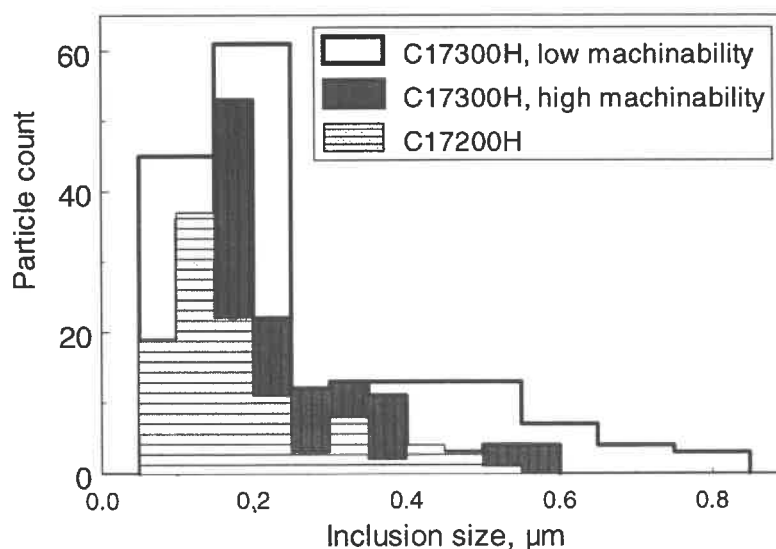


Figure 1. SEM picture of broken beryllides embedded in the chip.
Material C17300-HT, single crystalline diamond tool with -15° back rake, 20-80 nm edge sharpness, $5\text{ }\mu\text{m/rev}$ feedrate, $1\text{ }\mu\text{m}$ depth of cut.

Figure 2.
Distribution of beryllides and lead particles in Cu-Be samples. Images analyzed after scanning on SEM at 4000X.



The lead particles in the alloys lubricated the tool chip interface and slightly improved the surface finish. The surface finishes of the aged samples were slightly rougher than the unaged samples. This was probably due to the interference of the precipitates with the chip formation

mechanism. When viewing at 100x magnification, the visibility of Cu-Be grain boundaries on a micromachined surface indicated effective material removal, i.e., the materials were removed cleanly as chips rather than smearing on the surface.

Although a surface roughness measurement was consistent using one technique, the results varied up to 200-500% depending on different measuring techniques (Fig. 3). Different resolution, tip quality, and lack of a calibrating standard were among the reasons for such discrepancies. Because of this issue, measurement using one technique was used comparatively to evaluate the effect of process parameters.

Figure 3.

Surface finish of machined samples, measured with a scanning probe microscope (SPM), phase shift interferometer (PSI), and Form Talysurf (FT). Facing C17300-HT, single crystalline diamond tool.

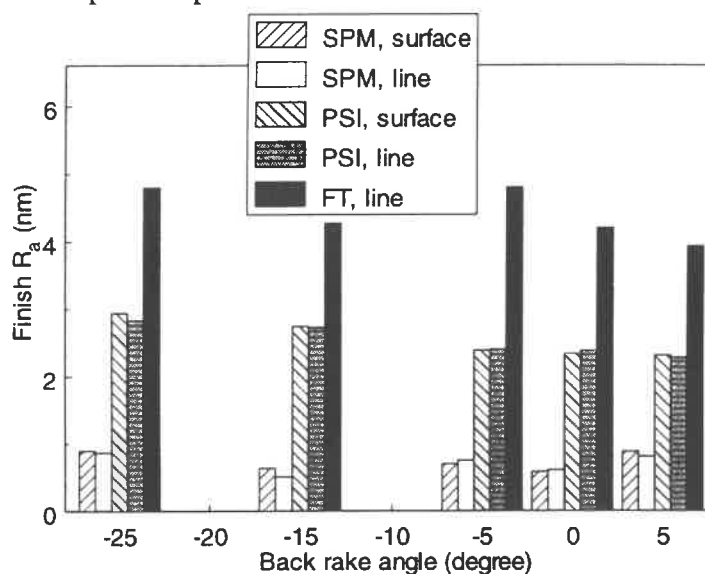
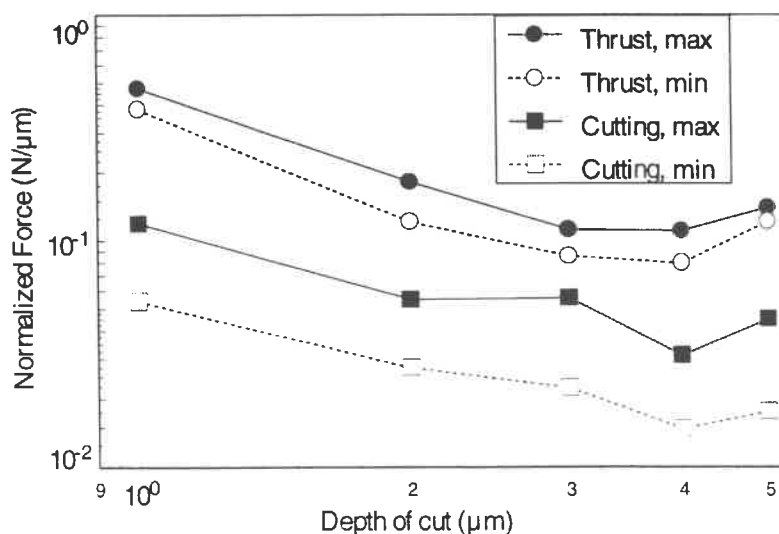


Figure 4.

Effect of depth of cut on cutting force (normalized to depth of cut). Material C17300, PCD tool, 0° back rake, 140 m/min, 2.5 μ m/rev feed.



Effect of machining parameters

As expected, the feedrate and tool nose radius significantly affected the surface finish. When ultraprecision machined with SCD tools, the best flatness of a samples was about 20 nm (RMS) when measured with an interferometer, and the best surface finish was about 1 nm R_a when measuring with a scanning probe microscope. An attempt to improve the surface finish further by lowering the ratio of feed/tool nose radius, unfortunately, was not successful.

The limit of surface finish was due to the smearing of beryllides during machining regardless of the selected machining parameters.

The depths of cut (0.2-1.0 μm) had insignificant impact on surface finish since they were much larger than the tool edge radius (sharpness) of 20-70 nm. The SCD tools with neutral or small positive back rake angles produced a better surface finish on both aged and unaged Cu-Be (Fig. 3). This was consistent with the published results for polymers after micromachining. As reported for other materials such as polymers and aluminum, the normalized cutting forces for Cu-Be also increased at shallow depths of cut and approaching the material bond strength (Fig. 4).

Polycrystalline diamond tools smeared the surface and produced a surface finish of 200-300% rougher than those produced by SCD tools. The smearing of material at the tool-work piece interface also increased the cutting forces. The force measurement showed a 100% increment of cutting forces due to PCD tools compared to those induced by SCD tools.

CONCLUSIONS AND RECOMMENDATIONS

Ultraprecision machining of Cu-Be alloys was investigated. This study showed:

1. A mirror finished surface of Cu-Be (20 nm flatness, 1 nm R_a and 8 nm R_t roughness) was obtained by microfacing. Visibility of grain boundaries on a machined surface confirmed the proper cutting conditions. This result was comparable with theoretical value, and was an improvement over the published data of 20-41 nm R_t for similar material.
2. Large beryllide inclusions worsened the micromachinability of Cu-Be since they chipped a fragile tool edge and plowed on the machined surface. Effort to reduce the size and distribution of beryllides was recommended.
3. Consistent measurement of surface finish was obtained, however, a large variation up to 500% was seen for different measurement techniques. There is a need for a standardized surface measuring technique, and surface references in the nanometric range.
4. The single crystalline diamond tool outperformed the polycrystalline tools in micromachining. The cost of the former, however, was a primary concern.

ACKNOWLEDGMENT

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Grinding Force Criteria in Shear-Mode Grinding

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1.Introduction

In shear mode grinding of brittle materials, the performance of grinding wheels is often described as the grinding force, which usually changes during the process. Therefore, many papers report qualitatively, that an extremely high grinding force is caused by shear mode grinding compared with brittle mode grinding. This corresponds with research results based on conventional grinding theory, i.e. 1) the profiles of cutting edges change during grinding, and 2) truing and dressing diversify the surface textures of wheels, all of which cause variation and transition of grinding force.

On the other hand, in our research, the constant grinding force was experimentally found in shear mode grinding of optical glass BK7[1]. In this case, a bronze bonded 3000 grit diamond wheel(straight type, Dia.100mm) was used to grind BK7 glass; an oil based grinding fluid was used.

However, at the set-up depth of cut of $12 \mu\text{m}$ (given stock removal rate $5.4 \text{ mm}^3/\text{min}$), the grinding force never reached a constant steady state. Therefore, we guess that the removal rate is limited by contact length of wheel to workpiece which might cause heat generation.

This work is to increase shear mode grinding efficiency by grinding at large depth of cut. This paper demonstrates that exciting the wheel at ultrasonic frequency to induce a coolant in the contact zone.

2. Shear mode grinding of BK7 glass using straight wheel

2.1 Metal bonded diamond wheel [Conventional wheel]

In our research, the constant grinding force was experimentally found in shear mode grinding of optical glass BK7 (*Figure 1*). Grinding forces were measured using a four-axis quartz piezoelectric measuring instrument (Type 9273 made by Kistler Co.). In this case, a 4mm wide bronze bonded 3000 grit diamond wheel(straight type, Dia.100mm) was used to grind 50mm long, 3mm wide samples of optical glass BK7; an oil based grinding fluid was used. The wheel was used on a grinding machine with a Professional Instrument's "Block Head" type 4R. Wheel speed was 1200~3000rpm and cross feed rate 50~500mm/min. The depth of cut was set up at $1 \sim 10 \mu\text{m}$ (given stock removal rates of $0.45 \sim 4.5 \text{ mm}^3/\text{min}$). The wheel had been previously used in a grinding experiment with more than 500 passes at the depth of cut of $2 \mu\text{m}$ after truing, and was used in this experiment without further truing and dressing.

As a result of experiments, it was shown that a) after a number of passes, the normal grinding force reached an approximate steady state, b) during grinding, the wheel surface roughness and cutting performance were invariant, c) wheel truing and dressing were not carried out, d) actual depth of cut was equal with setting depth of cut, e) surfaces characteristics of shear mode grinding, i.e. without pits and fractures, were produced, f) depth of subsurface damage was assumed to be almost uniform, and, g) other brittle materials, for example, Zerodur and Pyrex, showed the same results[2]. From these experimental results, shear mode grinding force criterion were defined as the steady, invariant grinding force which is a repeatable deterministic function of shear mode grinding conditions including grinding wheels and materials.

However, at the set-up depth of cut of $12 \mu\text{m}$ (given stock removal rate $5.4\text{mm}^3/\text{min}$), the grinding force never reached a constant steady state (*Figure 2*), and cracks caused by burning appeared on the ground surfaces; therefore, grinding was impossible. From this result, it can be surmised that the removal rate is limited by contact length of wheel to workpiece which might cause heat generation.

2.2 Serrate wheel

The 3000 grit diamond serrate wheel used for experiments are a W-1.0 wheel with a tooth width of 1mm (*Figure 3*). This wheel have a diameter of 100mm and a tooth thickness of 4mm, which are the same as conventional wheel. In the experiment, grinding conditions were as follows ; depth of cut $2 \mu\text{m}$, cross feed rate 150mm/min and wheel speed 1500rpm. As shown in *Figure 4*, grinding force was invariant, i.e. it showed the grinding force value lower than that obtained with conventional grinding wheel under the same grinding conditions. The grinding force ratio (F_t/F_n) of tangential grinding force (F_t) to normal grinding

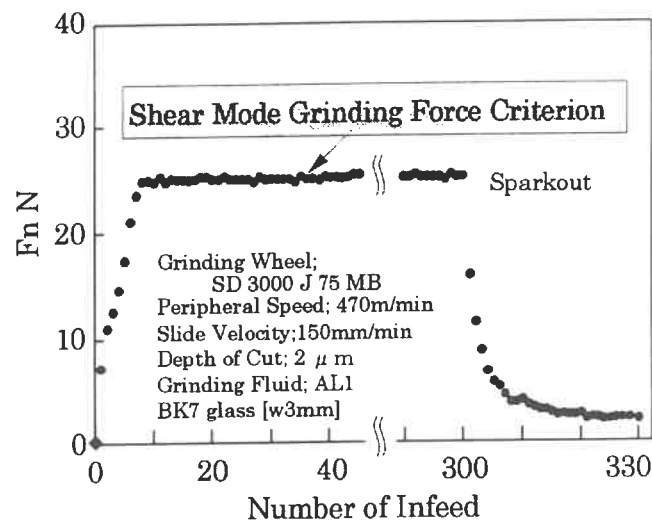


Fig. 1 Shear mode grinding force criterion.

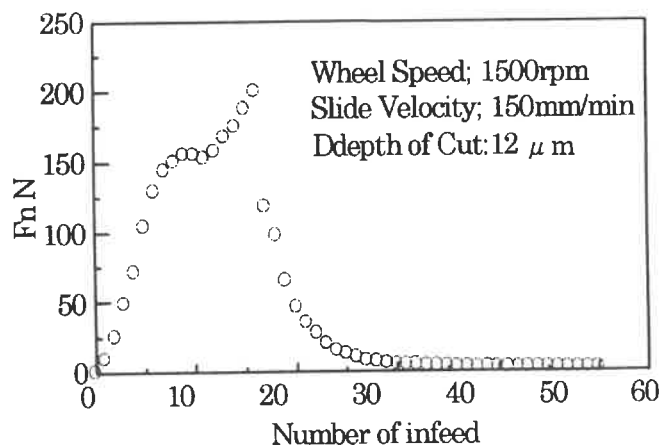


Fig.2 Change of normal grinding force of BK7.at $12 \mu\text{m}$

force (F_n) was found to have increased to 0.12 compared with 0.08 of conventional wheel. However, increasing the depth of cut to $4 \mu\text{m}$, the grinding force became instable, and at the depth of cut of $6 \mu\text{m}$, grinding burn was caused.

Removal rates per grain of conventional and serrate wheels were almost same, under the above mentioned grinding conditions before grinding burn occurs. From these results, it was assumed that serrate wheels facilitate cooling down of diamond grits by supplying more grinding fluid between the wheel and workpiece surface. This effect was also assumed to decrease the normal grinding force and increase the grinding force ratio as described above.

3 Shear mode grinding of BK7 glass using straight wheel with ultrasonic vibration

From above mentioned results, that the removal rate is limited by contact length of wheel to workpiece which might cause heat generation. The ability to perform high quality, shear mode grinding of brittle materials such as glass critically depends on an adequate flow of coolant through the contact zone between the tool and workpiece. A long contact zone limits the induction of coolant and promotes high temperatures in the coolant zone where heat is generated. For brittle materials like glass, the high temperature causes subsurface damage. For some grinding geometry, where a long contact length is difficult to avoid, an alternative is to excite the wheel at ultrasonic frequency, which increases amount of coolant between wheel and workpiece.

To overcome heat generation, we introduce 40kHz ultrasonic assisted shear mode grinding with a straight wheel, we assumed vibration may induce grinding fluid come into the contact zone. In our experiments, a metal bonded straight type wheel was used. The frequency of the ultrasonic vibration was 40kHz and the amplitude was $2 \sim 5 \mu\text{m}$ to the radius direction (Figure 5). BK7 glass was ground with ultrasonic vibration assisted grinding in shear mode. Wheel speed was

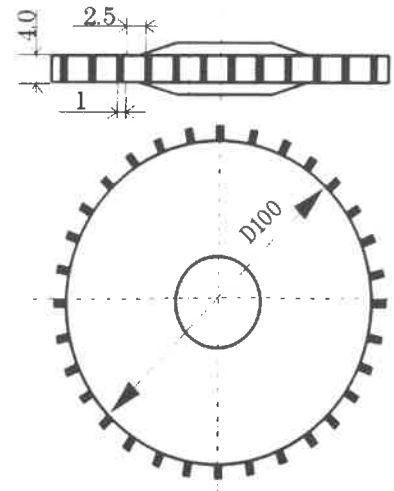


Fig.3 Schematic drawing of serrate grinding wheel

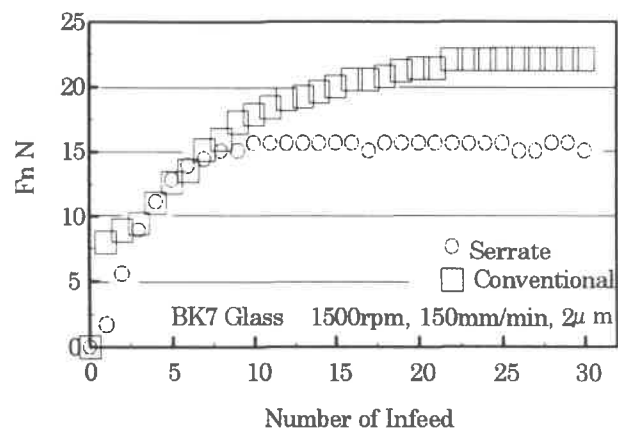


Fig.4 The change of normal grinding force using two different serrate wheels, compared with a conventional wheel

1500rpm and cross feed rate 150mm/min. The depth of cut was set up at $6\sim 20\ \mu\text{m}$ (given stock removal rates of $2.7\sim 9.0\text{mm}^3/\text{min}$).

The grinding force was almost constant at the set-up depth of cut of $12\ \mu\text{m}$ (Figure 6). A lower grinding force value than that of with non-ultrasonic vibration assisted shear mode grinding (usual grinding under the same conditions) was observed. Furthermore, exciting the wheel at ultrasonic frequency enhanced inducement of coolant in the contact zone, enabling an increase of the removal rate of BK7 glass to $9.0\text{mm}^3/\text{min}$ (depth of cut $20\ \mu\text{m}$). Consequently, 1) the removal rate increased two-fold, 2) the contact zone increased 1.4 times, 3) the removal rate per grain increased 1.5 times, compared with non-ultrasonic assisted vibration grinding.

4. Conclusion

We presumed that the removal rate was limited by contact length of wheel to workpiece causing heat generation. However, it was observed that exciting the wheel at 40kHz ultrasonic frequency further induced coolant in the contact zone. As a result of

experiments, it was found possible that removal rate of BK7 glass could be increased to $9.0\text{mm}^3/\text{min}$. This removal rate was two-fold, that of non-ultrasonic vibration assisted shear mode grinding.

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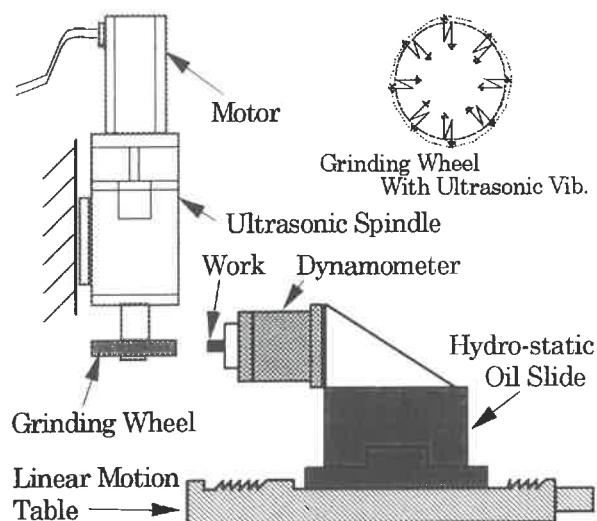


Fig.5 Schematic drawing of ultrasonic vibration assisted shear mode grinding machine

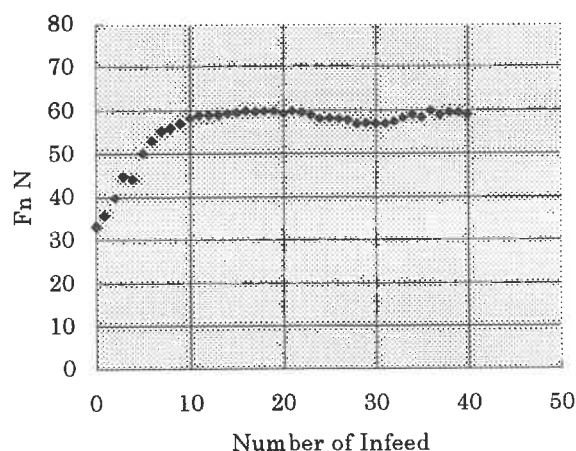


Fig.6 Change of normal grinding force of BK7 at $12\ \mu\text{m}$, with 40kHz ultrasonic vibration

INFLUENCE OF PHASE TRANSITION ON CUTTING DEPTH AND CHIP THICKNESS LIMITS IN DUCTILE MACHINING OF SILICON

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Extended Abstract

The ultraprecision diamond turning of single crystal semiconductors, (principally silicon and germanium) is an important sector of brittle materials machining research. The application of single point diamond turning to manufacture large lenses from semiconductor crystals is of considerable interest. Therefore, there are significant restrictions in machining these materials associated with hardness and brittleness properties. When a mechanical material removal process is applied to a single crystal, it is known that this will promote subsurface damage (Johansson & Shweitz 1989). Models based on fracture mechanics (Puttick & Franks, 1990, Hiatt & Strenkowski, 1991) and computer simulations through Molecular Dynamics (Shimada et al. 1995) have been proposed to explain the brittle-to-ductile transitions in the machining of these normally brittle materials. Recently, there has been some research taking into account the possibility that the ductility of such crystalline materials is related to a high pressure metallization (brittle-to-ductile) transformation occurring when they are subjected to high hydrostatic pressures and stresses (Morris et al. 1995). Because of the amorphous state observed in both chips and surface after the diamond turning of silicon (Shibata et al. 1994) and germanium (Morris et al. 1995), it is suggested that the results favour this hypothesis. In this work will be presented observations on the effect of phase transition on cutting depth and chip thickness in the ductile mode diamond turning of monocrystalline silicon. This will be done based on the measurements from the literature, of the thickness of the surface amorphous layer left by machining. The possibility will be discussed, based on SEM observations of chip morphology and topography, that chip formation might be a result of two mechanisms: shear and extrusion of material transformed between the tool-tip/edge and bulk material. It is suggested that the values of critical parameters such as critical chip thickness, obtained by different cutting conditions, might be related to the thickness of the layer undergoing a phase transition induced by the hydrostatic pressure combined with the high level of shear stress imposed by the cutting tool.

Results related to optimisation of the machining process, where economical cutting conditions and appropriate diamond tool geometries, have not yet reached complete agreement. The sequence of passes in order to obtain high and faster removal rates and achieve good surface roughness and surface and subsurface integrity in large lenses is not well known. For instance, it is well known, from the fabrication of aspheric surfaces in ductile metals, that the tool nose radius cannot be too large because of the necessity of tool compensation during the cutting path despite the good surface roughness level obtained with this tool geometry. Cutting conditions, such as maximum chip thickness and cutting depth, still remain without an accurately-known value for semiconductor single crystals. Some available information concerning the dimensions of the ductile region are presented for Si and Ge observed in Interrupted Cutting Tests (Blake & Scattergood, 1990, Blackley & Scattergood, 1991, and Tidwell & Scattergood, 1992). These results show that tool geometry and feed rate play a very relevant role in this phenomenon, but that there is no correlation between the nominal depth of cut and the brittle-to-ductile transition.

Recently, Shibata and his coworkers (1994,1996) demonstrated, through TEM analysis of the cross section of diamond turned silicon samples, that the subsurface damaged layers reached 2-3 μm deep into the workpiece for submicrometer depth of cut, but this value would decrease when the cutting depth is increased to micrometer range, although emerging microcracks in this case. However, with both cutting conditions, a 150nm amorphous layer on the surface, was always found. According to these workers, the mechanism of deformation responsible for the machining in the submicrometer range is the activation of slip systems and, for the micrometre range, there is an additional mechanism based on the generation and propagation of microcracks. Similar results were also obtained

by Puttick et al. (1994) and Jeynes et al. (1996). However, based on these findings, it could be asserted that if, after a cut made within the micrometre cut depth range, we proceed with another cutting pass but now in the submicrometre cut-depth range, small pits are likely to be observed on the surface due to this latter cut being made within the damaged layer, where there is a concentration of micro defects and microcracks. Figure 1 shows this effect on a (100) silicon sample cut with two passes, first with depth of cut of 5 μm and the subsequent with 0,5 μm . On the other hand, if the cut is made with depth of cut greater than the thickness of the damaged layer, the cut will now interact with the bulk material which is defect free, and it is likely that no pits will be seen on the surface. Another choice is to promote the subsequent pass within this expected value of the amorphous layer left on surface, which can be in the range of 100 - 250nm, as will be discussed in the next paragraph. So, it can be asserted that the surface integrity might be dependent on the sequence and dimensions of the cut-depths, based on the phase transition extent and the mechanisms of fracture related to crystal orientation.

On the other hand, the critical chip thickness is also another parameter that deserves consideration. Morris et al. (1995) mentioned that the observed morphology of germanium chips were virtually identical to that of soft metal chips. Meanwhile, it is also said that there is no evidence of dislocation activity in chips, and because of this, conventional plastic deformation modes were discarded. Instead, it was proposed that the amorphous phase that constitutes the chips was the likely end-result of a pressure induced phase transformation. However, as the metallic phase is reached during phase transition, the lamellae structure observed in the chip might be formed as a result of shear deformation. In this instance, the material should be extremely plastic, relative to the other two hard materials (Si and Diamond) which will be compressing this ductile mass. This would occur along with high hydrostatic pressure and shear stress, which make us believe that the lamellae formation of this chips might be also helped by extrusion of this transformed material as well as that firstly observed by Puttick and Hosseini (1980) and more recently by Phaer et al. (1991). This would possibly occur in a transformed field under the tool tip/edge and material interaction reminding "stress field" in indentation. But when the material reaches the end of this process, the lamella is formed and the chip is amorphous. This consideration is based on results obtained by Minowa & Sumino (1992) and Morris and Callahan (1994). In both, the authors demonstrated the formation of amorphous stress field below the tool path in scratching. Despite the difference in the thickness of this layer for the same load (2 g) condition, where the former was 150 nm and the latter 250 nm, it has to be mentioned that they carried out the tests on Si (100) and the latter on (111) surface orientation, respectively. Some clue on this discussion can be detected through the chips obtained shown in Figure 2 and 3. Figure 2 is a both side chip representation, where on the left side it is possible to observe the lamellae structure, and on the right side the face of the chip which contacts the tool rake face. Detail image of the back of the chip which is not smooth and mirror-like as it is observed in metal chips can be observe in Figure 3. On the contrary, it presents a kind of "scales" which could be termed as "lamellae tails", which seems to be a good clue to the extruding mechanism.

Based on the fact that, after the deformation process where phase transformation induced by pressure/stresses has taken place, an amorphous surface layer is formed and the values gained from the literature do not exceed a certain range, that is, 50 - 250 nm, it is proposed that this might be related to the critical chip thickness value which is the maximum thickness of the chip where the ductile-to-brittle transition occurred. Results presented from different mechanical processes such as indentation (Gridneva et al., 1972; Page et al., 1992), polishing (Johansson & Schweitz, 1989); diamond turning (Shibata et al. 1994,1996; Puttick et al. 1994; Jeynes et al. ,1996; and Inamura et al. 1997), demonstrated a similar range of values of the thickness of the amorphous layer left on the surface, e.g. 50 - 250 nm. On the other hand, the values of critical chip thickness measured by other workers are in the same range mentioned for the amorphous surface layer depth extension for silicon and germanium. For instance, 50 - 250nm for silicon and germanium is given by Blake and Sattergood (1990), 180nm for germanium by Tidwell and Scattergood (1991), 160-190nm for germanium by Blackley and Scattergood (1994) and 95 - 221nm for germanium and 57-91nm for silicon by Nakasuji et al. (1991). This means that the maximum value of ductile regime, observed along the uncut shoulder in Interrupted Cutting Tests, might be in agreement with the result of the maximum thickness of the layer that undergo phase transformation imposed by a combination of tool geometry and cutting conditions. Therefore, if this combination of cutting parameters exceeds the depth of the transformed layer at any point of the uncut shoulder, then the brittle mode will predominate from this point.

Concluding Remarks

Single-point diamond turning of monocrystalline silicon is a subject of great interest to researchers in ultra-precision machining and optical manufacture. Deeper understanding of the interaction between tool and material has been attracting the attention of researchers because of the potential explanation of ductility in silicon. The need of study the effect of this phenomenon on surface finish and integrity results from the manufacture of optical lenses from silicon and other related materials.

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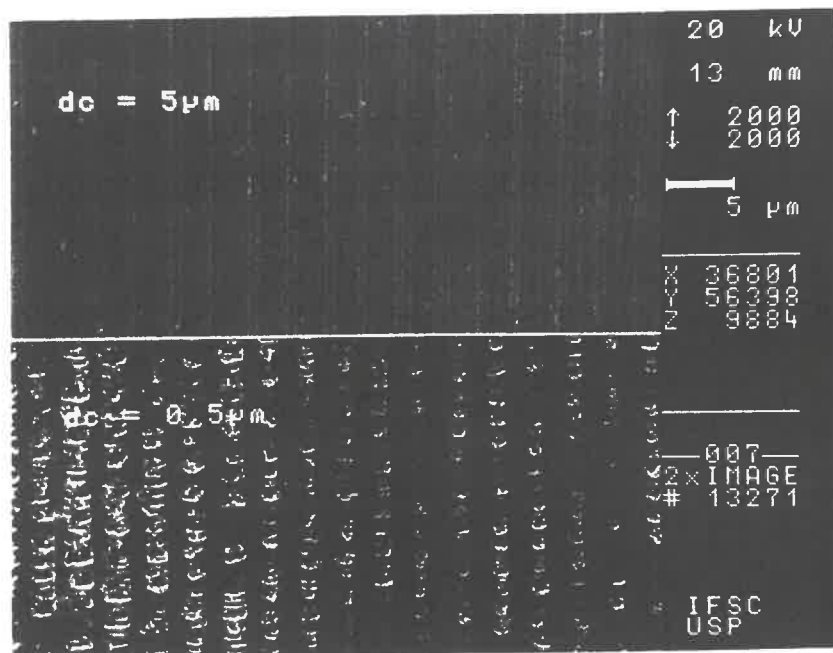


Figure 1. The effect of cutting depth sequence and dimensions on a (100) silicon sample cut with two passes, first with depth of cut of 5 μm and the subsequent with 0,5 μm .

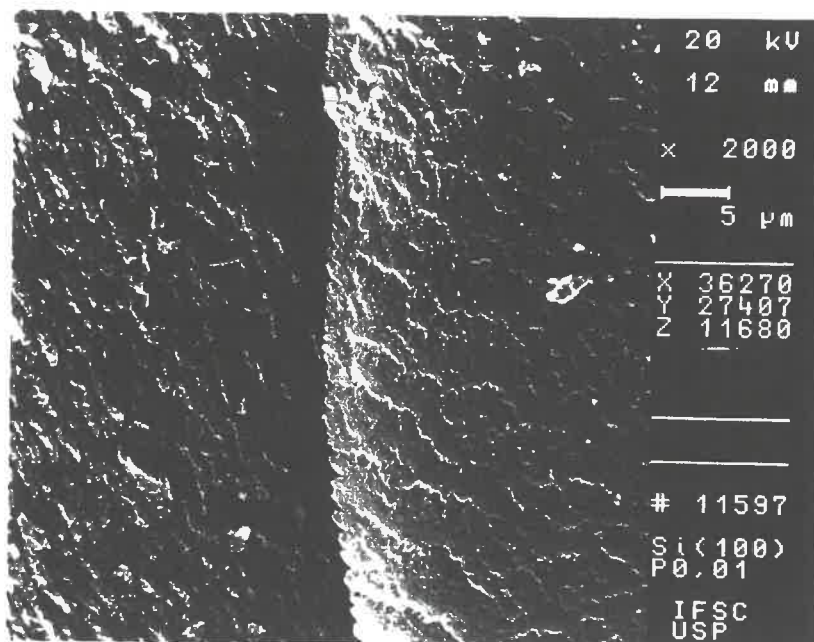


Figure 2 . Both side chip representation, where on the left side it is possible to observe the lamellae structure, and on the right side the face of the chip which contacts the tool rake face.

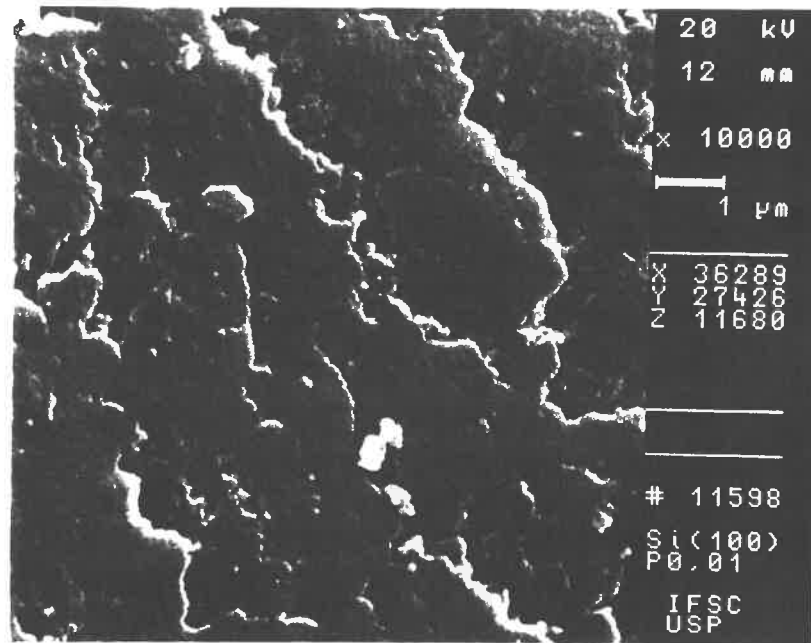


Figure 3 Detail of the back of the chip which is not smooth and mirror-like as it is observed in metal chips. On the contrary, it presents a kind of “scales” which could call as the “lamellas tails”, which seems to be a good clue of the extruding mechanism.

The Improvement of Form Accuracy by High Pressure Air Jet in Slot Grinding

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1. Abstract

Slot grinding is very efficient for the generation of micro ordered groove in hard and brittle materials. However, in continuous process, the form accuracy of the ground groove becomes worse by the wheel wear and loading phenomena. This paper focuses on the investigation of the effect of high pressure air jet on the grinding characteristics. Finally, we achieved a successful result that the change rate is 104% and 96% for slot width and depth respectively, after grinding of 10m distance with 800 μ m depth.

1. Introduction

One of the recent application in grinding technology is a slot grinding of optic and electronic parts with fine grooves⁽¹⁾. The material are too hard and brittle to grind, however, requires high form accuracy and fine surface roughness⁽²⁾. In conventional slot grinding using coolant, the ground depth becomes gradually shallow by wheel wear, the form accuracy becomes worse by increasing of ground slot width from the loading at lateral side of the wheel, resulting in chipping in workpiece. To prevent the loading phenomena of the wheel and improve the form accuracy of the workpiece, the high pressure air jet method is applied on the slot grinding and is compared with the coolant method.

2. Function of High Pressure Air Jet

During the removal of the workpiece surface by slot grinding, the wheel grains destroy tungsten carbide crystal on workpiece. Grinding fragments resemble fine powders rather than chips, which can easily get attached to the wheel surface causing a loading. In order to prevent this phenomena, high pressure air jet was used to remove impurities on the wheel surface. Fig.1 shows the effect of high pressure air jet on the wheel surface.

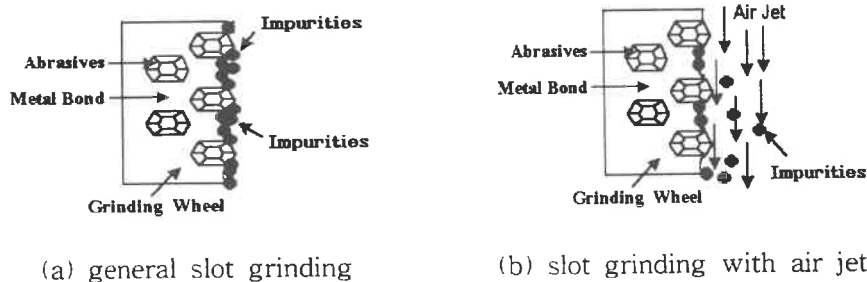
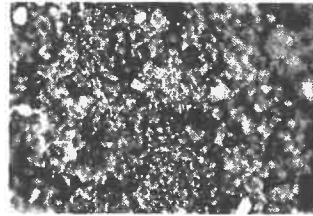


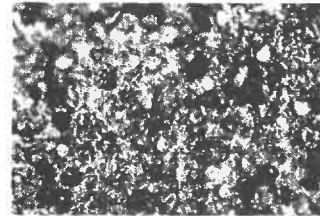
Fig.1 Effect of air jet

Fig 1 (a) shows conventional grinding process in which impurities stick to wheel surface thereby resulting in a loading phenomena, while (b) shows the use of high pressure air jet which removes impurities and prevents it. Fig.2 is a micrograph of the wheel surface after

grinding. Fig.2 (a) is the wheel surface after conventional grinding. The protrusion of wheel abrasives is not clearly visible and the wheel surface is in poor condition. Fig.2 (b) is the wheel surface when high pressure air jet is used to remove impurities. The protrusion of abrasives can be easily seen and the wheel surface is clear.



(a) without air jet

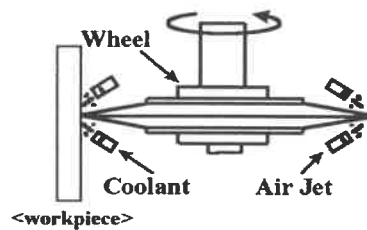


(b) with air jet

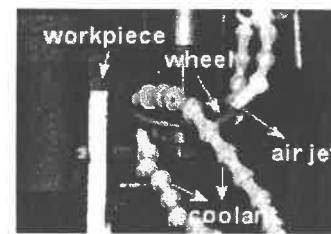
Fig.2 Micrograph of wheel surface after grinding with and without air jet

3. Experimental Procedure

Fig.3 is a schematic of the apparatus that uses high pressure air jet spray designed for this experiment. Air jet is sprayed to both side of the wheel surface in opposite position of workpiece. Experimental conditions are given in Table 1.



(a) schematic diagram



(b) experimental setup

Fig.3 Schematic and view of slot grinding with air jet

Table 1 Experimental conditions

Wheel	Cast iron diamond wheel ($\Phi 80$, 52° V shape, #270)
Grinding method	Slot grinding, down cut
Wheel speed	3000rpm($v = 760\text{m/min}$)
Work speed	100mm/min
Depth of cut	$t = 800\mu\text{m}$
Grinding fluid	Soluble type($\times 30$ diluted by water)
Workpiece	WC(M50), size : $10^L \times 10^W \times 5^l$
Machine	Vertical machining center(ACE V30)

Before the grinding, the wheel was dressed with an alumina stick to remove impurities between wheel grains. To measure and analyze the effects of the high pressure air jet, there are no dressing or truing during the grinding. The total grinding distance is 750cm($10\text{cm} \times 75$ passes). The machining width and depth were measured with a microscope (Olympus, MMDC201).

4. Results and Discussions

The experiment was performed under three conditions to investigate the effect on the form accuracy by high pressure air. Namely, 1) with coolant, 2) with high pressure air jet spray to the same direction and 3) the opposite direction as wheel rotation.

4-1 Conventional slot grinding (without air jet)

Fig.4 shows the V-face groove of the workpiece for various grinding passes after slot grinding, and Fig.5 compares the groove form after the first(bold line) and last pass(dashed line).

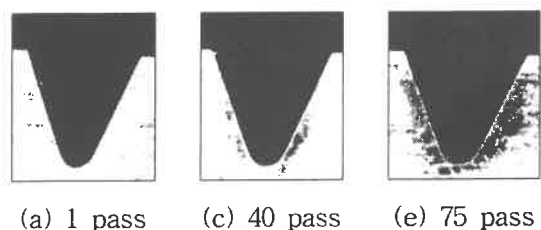


Fig.4 Cross sectional changes of V-Face Groove ground using Coolant



Fig.5 Ground form after first and last pass

After continuous grinding of 750cm distance, there was a decrease in depth of approximately $36\mu\text{m}$ and an increase in width of around $34\mu\text{m}$. Due to these changes, the slot angle was also changed with 4 degrees. The increase in width is due to the fine powder generated the workpiece which gets attached chemically and mechanically to the wheel surface and creates a loading phenomena. The fine powder causes chipping and worse surface quality of the groove. In addition, the powder has a negative effect on the groove's form accuracy.

4-2 Slot grinding with air jet (the same direction)

Fig.6 shows the V-face groove of the ground workpiece when high pressure air jet is sprayed in the same direction as the wheel rotation. Fig.7 is a quantitative comparison of the groove form after the first(bold line) and the last(dashed line) pass.

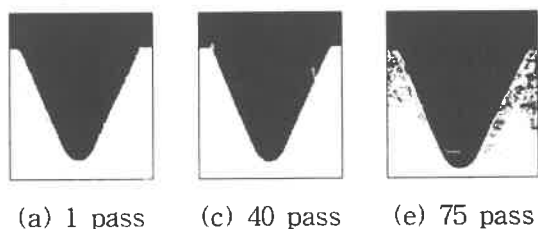


Fig.6 Cross sectional change of V-Face Groove ground using air jet (in same direction)

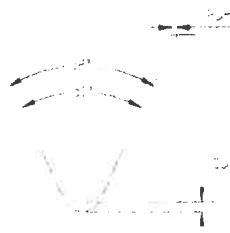


Fig.7 Ground form after first and last pass

The change in ground depth and width were approximately $10\mu\text{m}$ and $4\mu\text{m}$ respectively. 1 degree change of the ground groove angle shows that the groove form was maintained constant. This result indicates that the high pressure air jet removes the fine powder, and prevents the loading phenomena. This contribute the groove to maintaining a constant form.

4-3 Slot grinding with air jet (in the opposite direction)

Fig.8 shows the V-face groove of the ground workpiece when high pressure air jet is sprayed in the opposite direction as the wheel rotation. Fig.9 is a quantitative comparison of the groove

form after the first(bold line) and last(dashed line) pass.

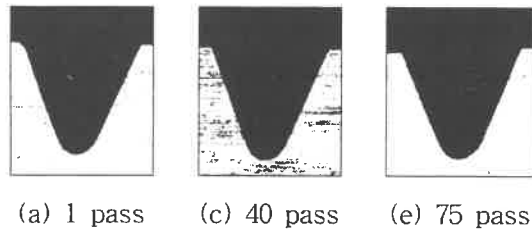


Fig.8 Cross sectional change of V-Face Groove ground using air jet (in opposite direction)

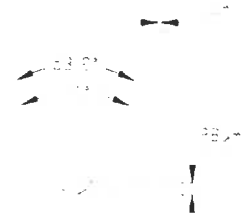


Fig.8 Ground form after first and last pass

The change of ground depth and width were approximately $28\mu\text{m}$ and $10\mu\text{m}$, respectively. The change of ground groove angle was a 2.2° . This result is better than that of when a coolant is mainly used.

4-4 Comparison of three experiments

Fig.10 and Fig.11 is a graph comparing the changes of ground width and depth obtained from the above experiments, respectively. The width and depth variations are relative comparison with a values obtained after first pass.

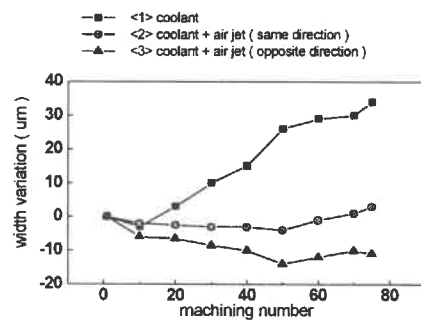


Fig.10 Width variation of ground slot

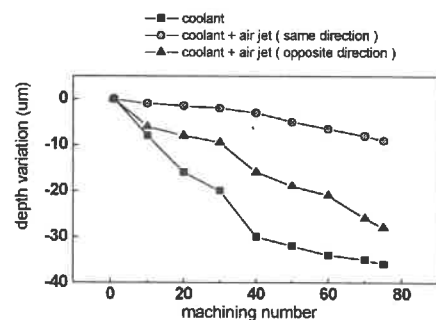


Fig.11 Depth variation of ground slot

As shown in Fig.10 and Fig.11, the variation of the width and depth of the ground slot was the smallest when high pressure air jet was sprayed in the same direction as the wheel rotation.

5. Conclusion

1. With a coolant, ground depth decreases due to wheel wear. While the width increases because of the loading phenomena on the both side of the wheel.
2. Form variations were smallest when high pressure air jet was sprayed in the same direction as the wheel rotation. Spraying in the opposition direction resulted on chattering in the wheel which deteriorated the form accuracy of the ground slot.
3. The decrease of the wheel wear rate, improvement of form accuracy, and chipping free characteristics obtained by using of the high pressure air jet.

6. Reference

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Precision Grinding of Micro Aspherical Surface - Feasibility Study of 250 μm Aspherical Grinding -

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1. Introduction

Recently, advanced optical fiber transmission equipments are being extensively used in various regions and the optical devices are becoming more and more small. Accordingly the micro precise aspherical molding dies have been increasingly required for the micro glass lenses. The sizes of the smallest lens are about 0.3 mm in diameter and the approximate radius curvature. The lens materials are glasses and manufactured by the glass molding method. The workpiece materials of the glass molding dies are ceramics such as WC (tungsten carbide) or SiC (silicon carbide). Therefore they must be ground ultra-precisely by diamond wheel¹⁾. In the previous report, a new grinding method and a system with the inclined rotational axis of micro grinding wheel were developed and the wheel was formed to a precise columnar shape of 0.8 mm in the diameter by a single crystal diamond truer on the machine. By the developed system, in grinding test of the workpiece of about 0.7 mm radius curvature, the form accuracy of about 0.1 μm P-V and surface roughness of less than 0.03 μm Ry were obtained²⁾. Recent trend is the further micronization of aspherical optical components. Therefore in this report micronizing of aspherical surface was examined by the high speed grinding spindle of 15×10^4 rpm at the maximum rotational rate and micro wheel of about 0.3 mm in radius curvature. In the grinding test, WC (tungsten carbide) molding die of about 0.25 mm radius curvature was tested as shown in Fig.1.

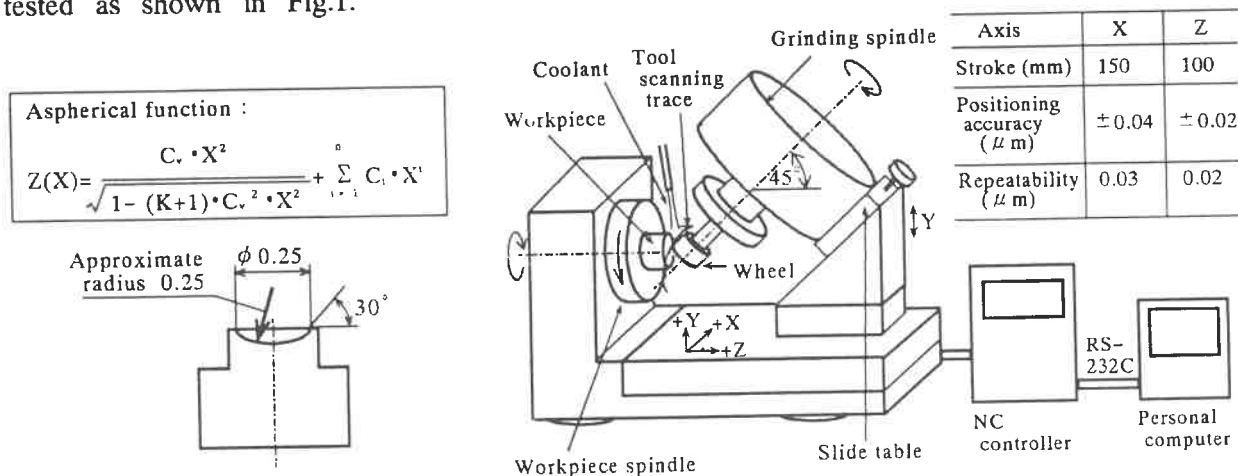


Fig.1 Shape of a target aspherical molding die

Fig.2 Schematic diagram of micro aspherical grinding machine

2. Grinding system of micro aspherical surface

2.1 Grinding machine

A new grinding method and a grinding system were developed for the micro aspherical surfaces. Fig.2 shows a schematic diagram of the micro aspherical grinding machine. The grinding wheel axis was 45 degrees inclined from workpiece rotational axis. The micro wheel was resinoid bonded diamond wheel and the wheel diameter was very small. The wheel was formed to a precise symmetrical column by a cup type of GC micro wheel on the machine. The grinding spindle was equipped with an air bearing and the maximum rotational rate of grinding wheel was 15×10^4 rpm. The grinding head was actuated by the simultaneous 2-axes (X,Z) linear scale feedback system. The wheel height was adjusted with Y-axis table.

2.2 Truing method of the micro wheel

Fig.3 shows a shape of resinoid bonded diamond wheel manufactured for trial in this experiment. The wheel was fixed at the grinding spindle by a collet chuck. The resinoid bonded diamond of #1200 was adhered in the shaft end surface and the pointed head was trued to be columnar of $\phi 0.3$ mm diameter. The micro wheel was trued with a cup type of GC truer fixed beside the workpiece spindle as shown in Fig.4. The truer rotational axis was fixed at the point of $D/2^{1.5}$ under the workpiece rotational axis, the wheel side surface was trued with the truer end surface and the wheel end surface was trued with the truer side as shown in Fig.4.

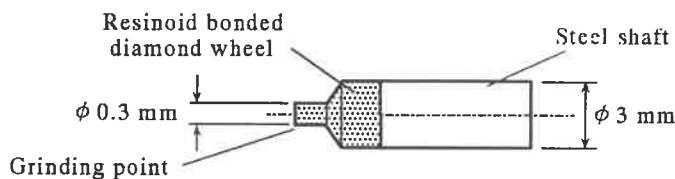
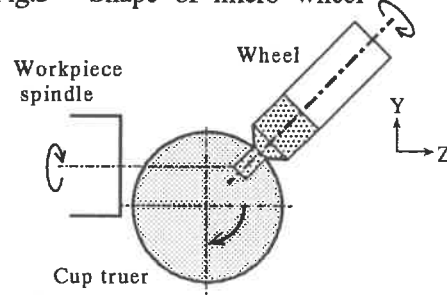
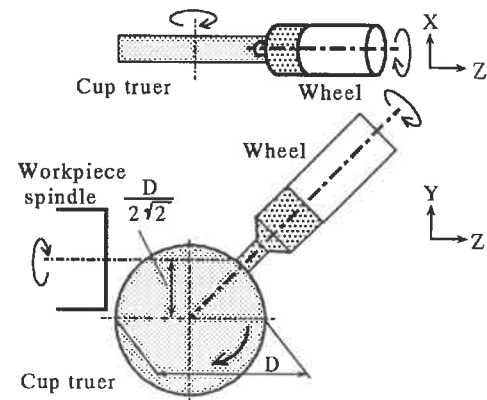


Fig.3 Shape of micro wheel



(a) Truing of side surface



(b) Truing of end surface

Fig.4 Truing method of micro wheel

Table 1 Truing conditions

Truer Grain size Rotational rate	GC cup type wheel #200 500 rpm
Wheel Rotational rate	#1200 diamond 1.5×10^4 rpm
Coolant	Solution type
Infeed rate	$0.5 \mu\text{m/rev}$



(a) Whole view

(b) Wheel edge

Fig.5 SEM photographs of the wheel

Table 1 shows truing conditions and Fig.5 shows the SEM photographs of trued wheel.

2.3 Grinding process

The wheel centroids (X_q, Y_q, Z_q) can be obtained from the coordinate of grinding point (X_g, Y_g, Z_g) and the normal vector $\vec{n}$ at the grinding point as shown in Fig.6 by the following formulae.

$$\left. \begin{aligned} X_q &= X_g + \frac{a}{l} \cdot r + \frac{a}{L} \cdot R \\ Y_q &= Y_g + \frac{b}{l} \cdot r + \frac{b+c}{2L} \cdot R \\ Z_q &= Z_g + \frac{c}{l} \cdot r + \frac{b+c}{2L} \cdot R \end{aligned} \right\} \quad (1)$$

where, $l = \sqrt{a^2 + b^2 + c^2}$ and $L = \sqrt{a^2 + (b+c)^2}/2$. Moreover, this machine is actuated by 2-axes (X,Z) drives, the centroids (X_q, Z_q) which realize $Y_q(X_g=0) = -R/\sqrt{2}$ are calculated numerically by using Newton-Raphson method.

Fig.7 shows the grinding process.

The tool profiles (outside diameter and tip radius) were at first measured to prepare an NC-program for the primary grinding. The primary ground surface was measured for the distribution of the form deviation profiles. This was to evaluate a reproducible errors such as the errors in tool wear. The NC-program was then edited to compensate the tool positioning coordinate (X_q, Z_q) with regard to the deviations of the tool positioning and the tool radius change with tool wear.

3. Grinding experiments

As a workpiece, the molding die shown in Fig.1 was tested and the material was tungsten carbide of ultra-fine particles.

3.1 Grinding conditions

The grinding conditions are shown in Table 2. As a wheel, a resinoid bonded diamond wheel of #1200 in a grain size was used. The wheel shape was columnar and the tip end was formed sharply as shown in Fig.3. In Eq.(1), the initial tip radius of the wheel r was left to be 0 and it increased as the wheel wore.

Table 2 Grinding conditions

Wheel	Resinoid bonded diamond
Grain size	#1200
Diameter	$\phi 275 \mu\text{m}$
Rotational rate	$12 \times 10^4 \text{ rpm}$
Depth of cut	$0.5 \mu\text{m}$
Coolant	Solution type
Feed rate	$0.05 \mu\text{m/rev}$

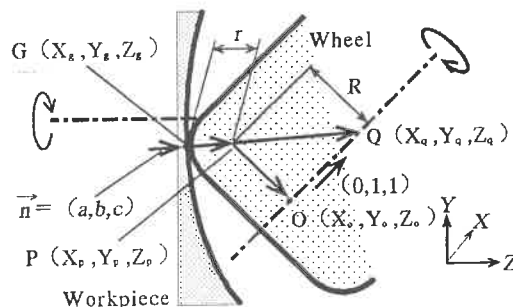


Fig.6 Calculations of tool path

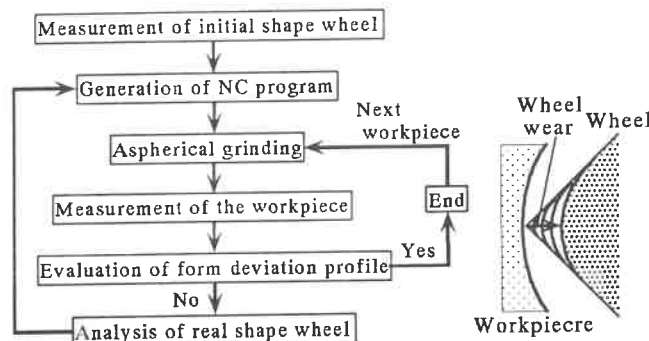


Fig.7 Grinding process

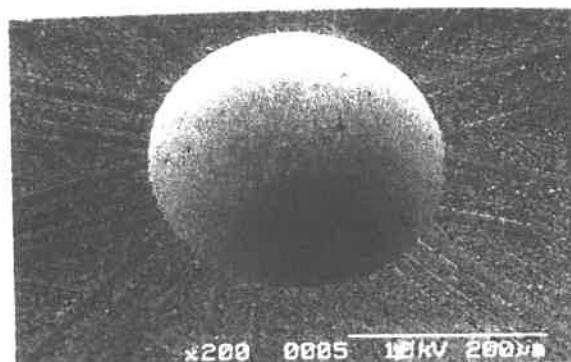


Fig.8 SEM photograph of ground workpiece

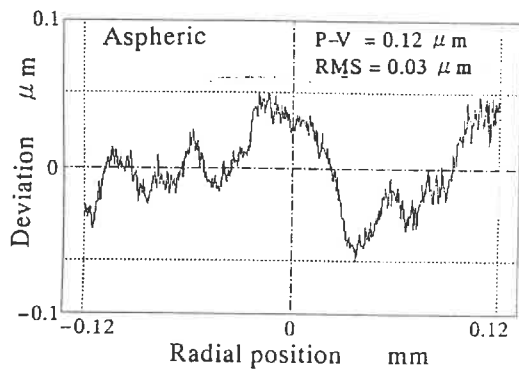


Fig.9 Form deviation profile after final grinding



Fig.11 Nomarski micrograph of ground surface

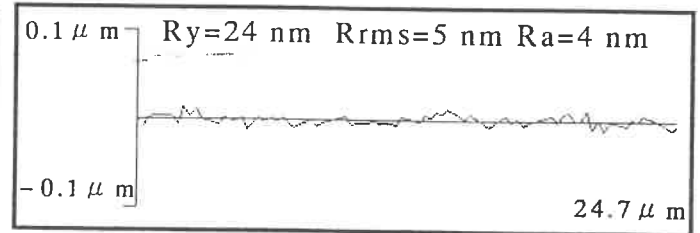
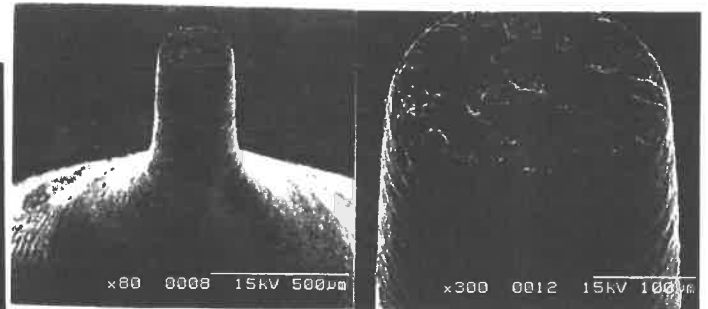


Fig.10 Surface roughness profile after grinding



(a) Whole view (b) Wheel edge

Fig.12 SEM photographs of micro wheel after grinding

3.2 Grinding results

Fig.8 shows a SEM photograph of ground surface. Fig.9 shows the deviation profile after final grinding. The form deviation profile was measured with Form Talysurf. The form accuracy was about $0.1 \mu\text{m}$ P-V. Fig.10 shows the surface roughness profile after grinding. The surface roughness of less than $0.03 \mu\text{m}$ R_y was obtained. Fig.11 shows a Nomarski micrograph of the ground surface. The ductile mode surface was obtained. Fig.12 shows the SEM photographs of the wheel after grinding. Fig.13 shows the change of the wheel wear with grinding pass in case of aspherical grinding as shown in Table 2.

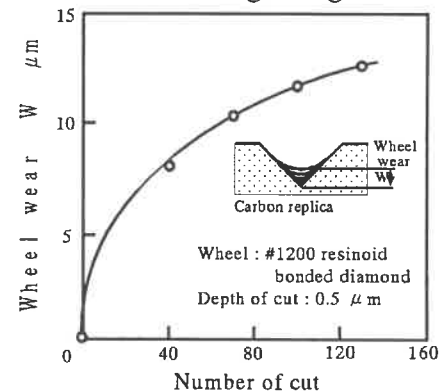


Fig.13 Change of the micro wheel wear with grinding pass

4. Conclusions

A High speed inclined rotational grinding system, micro wheel and its truing system were developed and a feasibility study of micro grinding with $\phi 250 \mu\text{m}$ aspherical surface was carried out. From the experiment, it was clarified that ultra-precision grinding can be performed with a resinoid bonded diamond wheel.

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Elliptical Vibration Grinding with an Ultrasonic Motor

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1. Introduction

There have been various trials of utilizing ultrasonic vibration in cutting or grinding hard-to-machine materials in expectation of reducing machining force, improving machining efficiency, restraining tool wear and restraining loading on wheels and burrs on workpieces[1][2]. In ultrasonic grinding, longitudinal vibration and bending vibration have mainly been used. On the other hand, the authors showed that it is possible to grind optical glasses and ceramics with a non-rotational tool by utilizing ultrasonic longitudinal and torsional vibrations of the tool[3]. Recently a unique cutting method utilizing elliptical vibration of a cutting tool tip was developed by Syamoto and Moriwaki[4]. In this method, exclusion of removed chips is promoted by the turn over of friction force on a rake surface of a cutting tip, and cutting force, tip temperature and burrs are remarkably reduced.

In this paper, an ultrasonic elliptical vibration grinding wheel was developed by plating diamond grits on a stator of an ultrasonic motor in which a rotor is rotated by elliptical vibration of a stator, and the effect of the elliptical vibration on grinding performance was investigated.

2. Development of an ultrasonic elliptical vibration grinding wheel

2.1 Application of a stator of an ultrasonic motor to a grinding wheel

The surface of a stator of an ultrasonic motor is vibrated elliptically by piezoelectric elements pasted on the back side of the stator as shown in Fig.1 and progressive wave is generated on the stator surface for rotating the rotor of the ultrasonic motor. The authors propose to produce a cup type ultrasonic elliptical vibration grinding wheel whose grits vibrate elliptically, by fixing abrasive grits on the surface of the stator by plating or some other methods as shown in Fig.2.

2.2 Motion of abrasive grits

Figure 3 shows trajectories of abrasive grits vibrating elliptically on the surface of a rotating grinding wheel proposed here and grits vibrating in conventional longitudinal vibration mode. In case of the longitudinal vibration mode, the trajectory of grits is sine wave. In case of the elliptical vibration

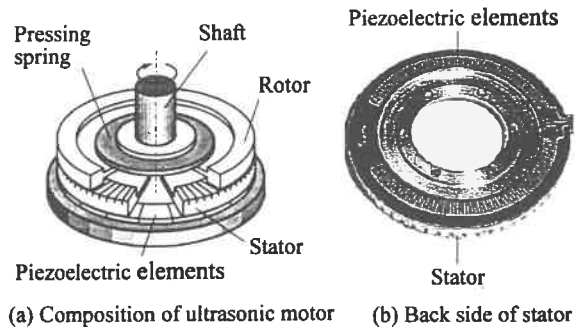


Fig.1 Structure of ultrasonic motor

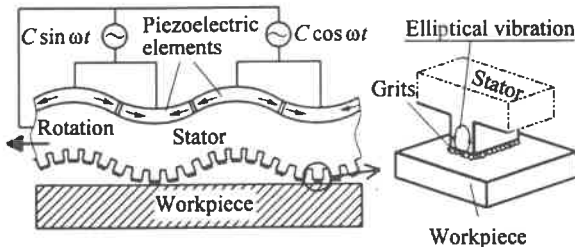


Fig.2 Schematic principle of ultrasonic elliptical vibration grinding

Vibration modes ($f=35\text{kHz}$, $\delta=2\mu\text{m}$)	Wheel rotation direction ($V_s=10\text{m/min}$)
Non-vibration	
Longitudinal vibration	
Elliptical vibration (acceleration vibration)	
Elliptical vibration (deceleration vibration)	

Fig.3 Simulation of abrasive grit trajectory in ultrasonic vibration grinding

Table 1 Expected features of ultrasonic elliptical vibration grinding wheel

[Expected effects in grinding]	
· Improvement of grinding efficiency	
· Promotion of exclusion of chips (restraint of loading)	
· Improvement of tool life	
· Reduction of grinding force	
· Restraint of chipping	
· Restraint of burrs	
· Grinding at low wheel peripheral speed	
· Grinding at low pressure	
[Features of utilizing ultrasonic motor]	
· Realization of elliptical vibration	
· Reduction of wheel cost by using a stator	
· Realization of thin wheels	
· Freedom of selecting wheel diameter	
· Impossibility of exchanging only wheel	
· Low stiffness of wheel	

mode, there are two types of trajectory of grits. One is like a trochoid curve, when the directions of the wheel rotation and the elliptical vibration at grinding points are same (named acceleration vibration). The other is the reverse of the above curve when those directions are opposite to one another (named deceleration vibration). In case of the acceleration vibration, the grits penetrate the workpiece with a rather small angle, but in case of deceleration vibration, the penetration angle is very large and steep.

Though Fig.3 shows trajectories in case of a combination of wheel peripheral speed of $V_s=10\text{m/min}$, vibration frequency of $f=35\text{kHz}$ and vibration amplitude of $\delta=2\mu\text{m}$, the tendency of the trajectory is same for other combinations of grinding and vibration conditions.

2.3 Features of the ultrasonic elliptical vibration grinding wheel

By utilizing an ultrasonic motor as a grinding wheel, various features are expected as listed in Table 1. The features are classified into two groups of ① improving grinding characteristics by applying ultrasonic vibration and ② improving availability of ultrasonic grinding devices.

2.4 Production of the ultrasonic elliptical vibration grinding wheel

A cup type ultrasonic elliptical vibration grinding wheel was produced by plating diamond grits (SD1000) on the stator of a commercially available ultrasonic motor with a rather large outer diameter of $\phi 60\text{mm}$ and rated torque of $5.88 \times 10^{-2}\text{N} \cdot \text{m}$. Figure 4 shows the outlook of the wheel, size of a tooth of the wheel and a state of plated diamond grits on the stator. Before plating diamond grits, frictional stuff on the stator was removed and the tooth edges of the stator were rounded. Diamond grits were plated only on the surface of the tooth by

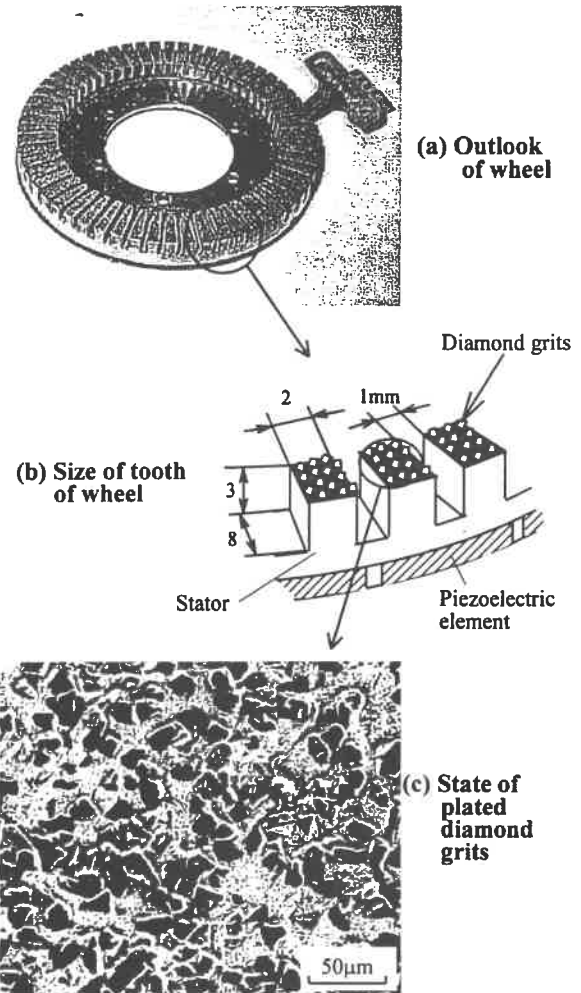


Fig.4 Prototype ultrasonic elliptical vibration grinding wheel

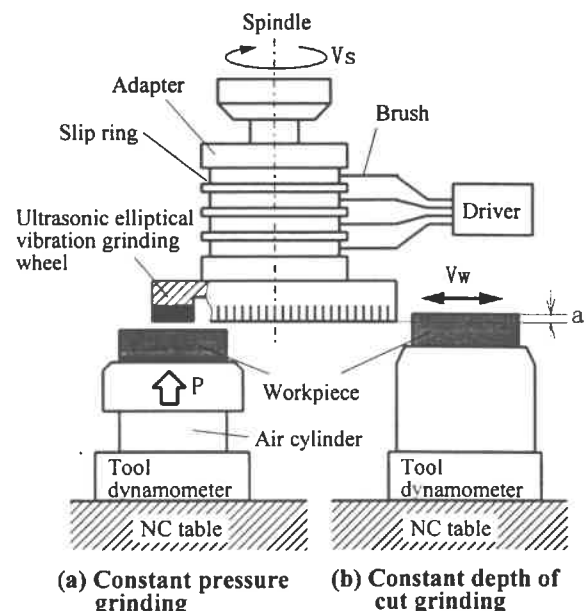


Fig.5 Schematic illustration of experimental setup

Table 2 Experimental devices and conditions

Machine	3-D NC machine(A35R,Sodick)
Ultrasonic elliptical vibration grinding wheel	Base motor:UA-60 (Canon) Diameter:Do/Di=60/44mm Teeth:2W*8L*3H, 58 pieces, Groove width:1mm Plated grit:SD1000 (Noritake Dia) Driver:UA60NIT (Canon)
Workpiece	Monocrystalline silicon(20×10mm, 31×40mm)
Vibration conditions	Direction:Acceleration, Deceleration Frequency:f=35kHz Amplitude: $\delta_x=\delta_y=2\mu\text{m-p}$ (Circular) Vibration speed:Vv=13.2m/min(0.22m/s)
Constant depth of cut grinding conditions	Vs=10,20,30,40 m/min Vw=100mm/min, 10passes a=1,2,3 μm Grinding fluid:JC707(dilution × 50, Castrol)
Constant pressure grinding conditions	Vs=10,20 40,60 m/min, Vw=0mm/min, L=0mm P=15,25kPa(F=4N), S=267mm ² , t=6min JC707×50 (Castrol)
Measuring instruments	Tool dynamometer (9257B,Kistler), Profilometer (Form Talysurf S-4C, Taylor Hobson), SEM (S-2380N,Hitachi)

masking.

It was confirmed that the grits of the produced wheel vibrate at a frequency of $f=35\text{kHz}$ with maximum longitudinal and lateral amplitudes of $\delta=2\mu\text{m-p}$ in nearly circular motion. The vibration speed was $Vv=13.2\text{m/min}$.

3. Experimental setup and conditions

Experiments of constant depth of cut grinding and constant pressure grinding for monocrystalline silicon workpieces were conducted with the produced ultrasonic elliptical vibration grinding wheel. Figure 5 shows the schematic experimental setup. The ultrasonic elliptical vibration grinding wheel was fixed to a spindle of a 3-D NC machine with an adapter. Electricity for driving the ultrasonic motor was supplied to the wheel through a slip ring on the adapter. Grinding forces were measured by a tool dynamometer. In case of the constant pressure grinding, the workpiece of silicon plate was fixed on a constant pressure equipment consisting of an air cylinder with a linear motion guide. A small quantity of grinding fluid was supplied to the wheel during grinding. Table 2 shows the experimental devices and conditions.

4. Results of grinding experiments

4.1 Constant depth of cut grinding

(1) Effects of wheel peripheral speed

Figure 6 shows relationship between normal grinding force F_n and wheel peripheral speed V_s in grinding with constant depth of cut of $a=2\mu\text{m}$. In any case of the wheel peripheral speed of 10 to 40m/min, the normal grinding force decreased remarkably by giving elliptical vibration to the wheel. The lower the wheel peripheral speed was, the more the effect of

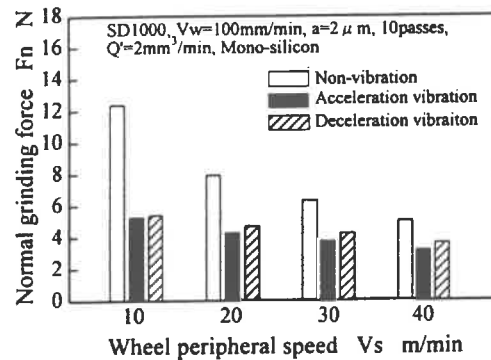


Fig.6 Relationship between grinding force and wheel peripheral speed (constant depth of cut grinding, $a=2\mu\text{m}$)

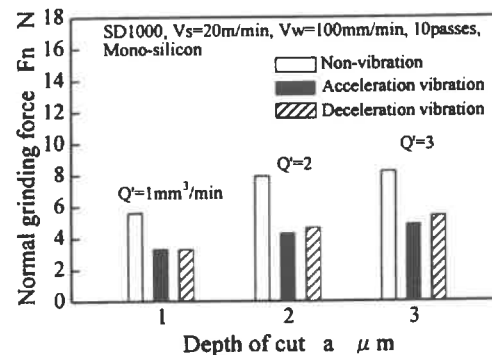


Fig.7 Relationship between grinding force and depth of cut (constant depth of cut grinding)

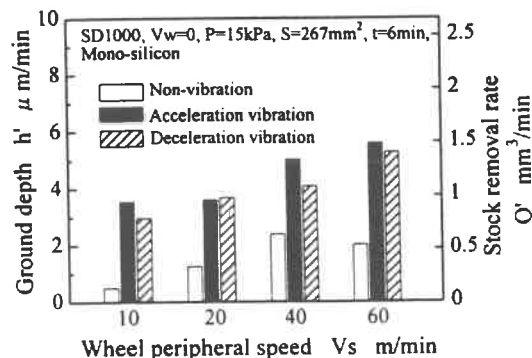


Fig.8 Relationship between stock removal rate and wheel peripheral speed (constant pressure grinding, $P=15\text{kPa}$)

elliptical vibration was. When the wheel peripheral speed was 10m/min, the normal grinding force in elliptical vibration grinding was nearly half comparing with non-vibration grinding. The normal grinding forces in case of acceleration vibration were a little lower than those in case of deceleration vibration.

(2) Effects of depth of cut

Figure 7 shows relationship between normal grinding force and depth of cut on the condition of wheel peripheral speed of $V_s=20\text{m/min}$. The normal grinding force decreased by 40% by giving elliptical

vibration in any case of the depth of cut of 1 to 3 μ m.

4.2 Constant pressure grinding

Figure 8 shows relationship between stock removal rate Q' (removed depth per minute : h') and wheel peripheral speed V_s at a constant and rather low grinding pressure of $P=15\text{kPa}$. The stock removal rate increased steeply by giving elliptical vibration to the wheel. The lower the wheel peripheral speed was, the more remarkable the increase of stock removal rate was. Comparing with non-vibration grinding, 8 times and 2.5 times stock removal rates were obtained in case of $V_s=10\text{m/min}$ and 60m/min , respectively. This result means that the ultrasonic elliptical vibration grinding wheel is suitable for using under lower pressure and lower peripheral speed conditions. The stock removal rates in case of acceleration vibration were a little higher than those in case of deceleration vibration.

4.3 Conditions of ground surface

There was little difference of surface roughness on any conditions of wheel peripheral speed, depth of cut, grinding pressure and with or without vibration, and the surface roughness was $R_{z_{ISO}}=4\text{--}5\mu\text{m}$. But there occurred some difference of ground surface conditions by giving elliptical vibration to the wheel. Figure 10 shows SEM pictures of ground surfaces. In case of non-vibration mode, large cracks were observed everywhere on the ground surface. But in case of vibration modes, few cracks were observed on the ground surface and the surface was formed by micro destructions.

5. Conclusions

By plating diamond grits on a stator of an ultrasonic motor, a cup type ultrasonic elliptical vibration grinding wheel, each of whose diamond grits vibrates elliptically during grinding, was successfully developed. Grinding experiments for monocrystalline silicon workpieces showed that normal grinding force decreased steeply to 50% in constant depth of cut grinding by giving elliptical vibration and that 8 times stock removal rate was obtained in constant pressure grinding. The effects of elliptical vibration were remarkable in case of rather low wheel peripheral speed as 10m/min and low grinding pressure as 15kPa . The ground surface was observed to be with few large cracks and be formed by micro destructions.

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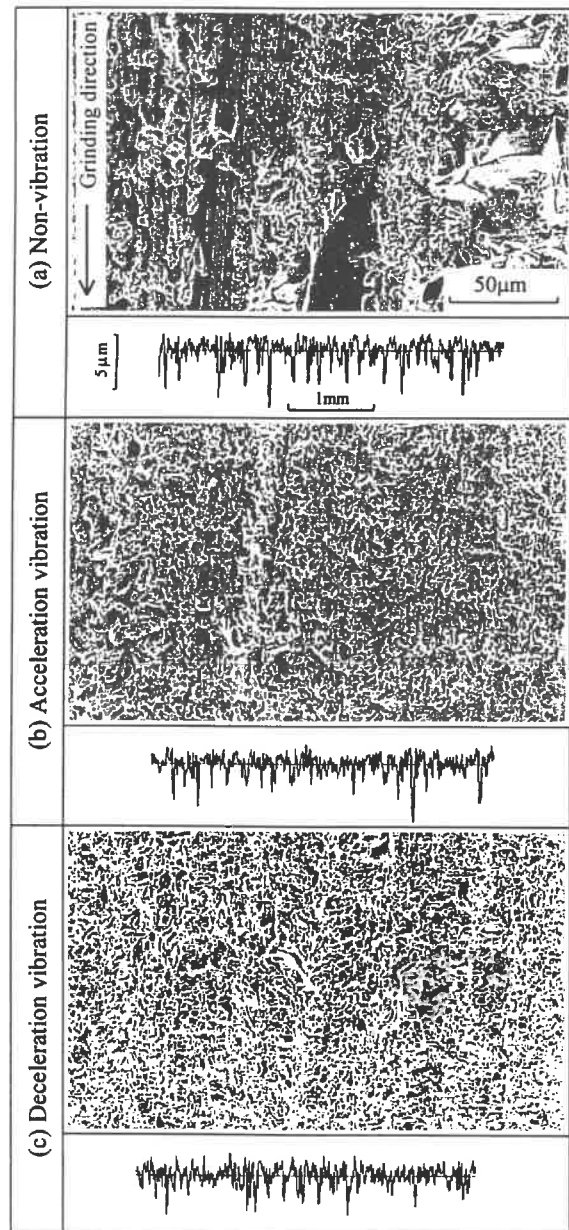


Fig.9 Conditions of ground surface
($V_s=10\text{m/min}$, $P=25\text{kPa}=\text{const.}$)

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Rapid Prototyping—Its New Possibility of Manufacturing of Grinding Wheel

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The present paper proposes the unique technology that develop grinding wheel by curing and layering the ultra violet curing resin which is mixed abrasive grain. The grinding characteristics of rapid prototyping wheel are analyzed in grinding tests. It is cleared that the cured depth and width of the ultra violet curing resin increase with the long exposed hour and only 5 to 10 sec exposure of ultra violet is eligible to fabricate a small area of grinding wheel surface. It is possible to manufacture the polishing disk and grinding wheel which hold tightly and distribute uniformly abrasive grain. The grinding ability of rapid prototyping wheel is large due to the enough bending strength and elastic modulus.

INTRODUCTION

Rapid prototyping is the unique technology which makes the 3-d model with the resin cured by ultra violet. It is the fabricating technology that forms precisely the desired model with the layered resin of a local solid transformed from liquid. Many procedures which build up directly the various models and micro machine parts have been developed and tried.

The creative idea bases on the apply of the rapid prototyping principal and technology which the resin bonded grinding wheels (RP wheel) are composed with the stepped layers of the ultra violet curing resin mixed with abrasive grain. It is strongly expected that not only abrasive grains may be uniformly distributed in the liquid resin, but also it is possible to manufacture the form grinding wheels of complex shape. Furthermore, it seems that the very soft and hard wheels, the tough and elastic wheels and the pore type wheel will be manufactured with such resins.

Thus, the present paper mainly describes the curing characteristics of resins, the manufacturing method of polishing disk and grinding wheel, and the verification of their machining abilities by the polishing and grinding tests. The fundamental examinations are the investigation of the curing

characteristics of the abrasive grain mixed resin and the ability of polishing disk and grinding wheel is discussed in the applied test.

RESIN AND EXPERIMENTAL PROCEDURES

Table 1 The ultra violet curing resins

No	Viscosity cps/25°C	Tensile strength MPa	Elongation %
M-305	400~800	39	0~5
M-8030	500~1100	20~30	5~10
M-240	10~30	19	0~10
M-6100	200~500	50	10~30
M-6250	300~700	10	30
M-1210	2600~3200	5.5	35
M-113	50~130	0.1	50
M-117	70~150	0.3	60

ULTRA VIOLET CURING RESINS

Table 1 shows the characteristics of the ultra violet curing resins of acrylate. These eight resins have not any poison.

MIXING AND CURING OF RESINS

Abrasive grain is mixed in resin and they are stirred for 30 min with the 5 wt% cure. The prepared resin is

cured by ultra violet of 354 nm wave length.

POLISHING AND GRINDING EXPERIMENTS

The polishing disk is settled on the polishing machine. The workpiece is molybdenum which has Vickers hardness of 250, the tensile strength of 520 MPa and the elongation of 55 %.

Table 2 shows the grinding conditions. The grinding center was used for plunge grinding test. The working surface of wheel was observed with SEM. Also, the change of grain worn area was measured with the increase of grinding distance. The grain distance histogram was obtained from the average grain distance.

PROPERTIES OF GRAIN MIXED RESIN

Figure 1 shows the changes of the cured depth by the exposed time of ultra violet. The cured depth of grain mixed resin decreases with more alumina contents by their absorption and dispersion. But, no change of the cured depth is observed over 40wt% alumina content. The hidden side of grain will be cured by ultra violet permeated through alumina. Thus, the cured depth increases. However, the cured depth decreases with more alumina content, because the influence of dispersion and absorption of ultra violet is larger than that of permeation.

Figure 2 shows that the bending strength of the abrasive grain mixed resin increases with much alumina contents by the obstruction of crack transmission. The elastic modulus of resin itself is large, but the mixed resin shows the small elastic modulus.

MANUFACTURE OF WHEEL AND DISK

One of the resins is used to make a polishing disk (RP disk) of ϕ 150 under the ultra violet exposed time of 5 to 10 sec. One layered thickness of the abrasive grain mixed resin is 0.1~0.2 mm. The grain sizes are #400, #1000, #3000 and #8000.

On the other hand, a grinding wheel (RP wheel) has the size of ϕ 100×6, the mesh size of abrasive grain is #1000 and the volume percentage is about 25 vol%.

By SEM observation, the fracture cross area of bending test specimen does not indicate any stepped

Table 2 Grinding conditions

Grain	WA #1000
Workpiece	Molybdenum ϕ 20x10
Wheel speed	30 m/s
Table speed	0.5 mm/s
Wheel depth of cut	1 μ m
Concentration	100 (25 vol%)
Grinding liquid	Semicool

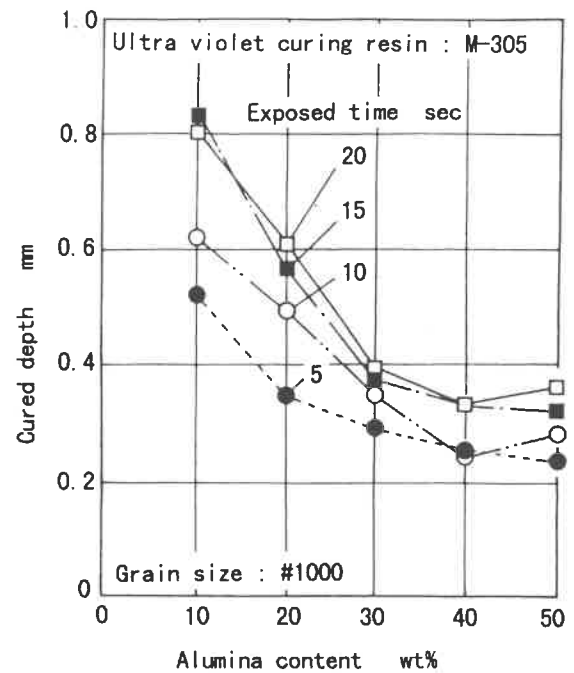


Figure 1 Cured depth of grain mixed resin

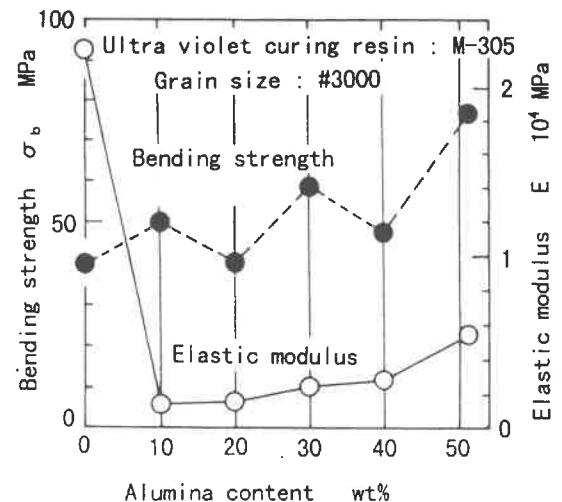


Figure 2 Mechanical properties of grain mixed resin

Table 3 Improvement of roughness by polishing

M-305	Ra (μm)	Ry (μm)
Pre-polish	0.074	0.353
After polish #400	0.04	0.238
After polish #1000	0.018	0.137
After polish #3000	0.009	0.072
After polish #8000	0.006	0.039

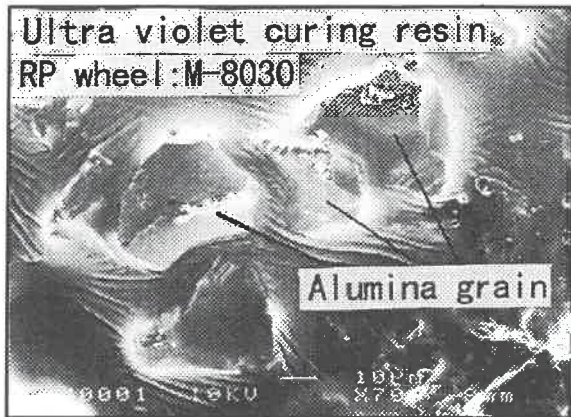


Figure 3 Adhesive conditions of grain with resin

interface layer. It seems that the interface disappears due to the resin cured at the hidden side of grain by the permeation of ultra violet. On the other hand, the resin adhere tightly to abrasive grain and there is no clearance, as shown in *Figure 3*.

MACHINING ABILITY OF DISK AND WHEEL

Polishing results were shown in *Table 3*. The polishing test shows the decrease of polished surface roughness from $0.353 \mu\text{mR}_y$ to $0.039 \mu\text{mR}_y$ with #8000 grits. The mirror surface was obtained on molybdenum polishing plane with the tiny scratching grooves. It is possible to polish a hard metal by RP disk.

On the other hand, the grinding ability of wheel may be presented by the changes of grain area ratio. *Figure 4* shows that the grain area ratio changes slightly in RP wheel compared with the generally used WA wheel, because of the little releasing of abrasive grains. *Figure 5* shows the worn surface of wheel. There are a lot of abrasive grains which are tightly hold in bond after dressing. With more grinding distance, the wheel working surface is going to be flat due to the abrasive

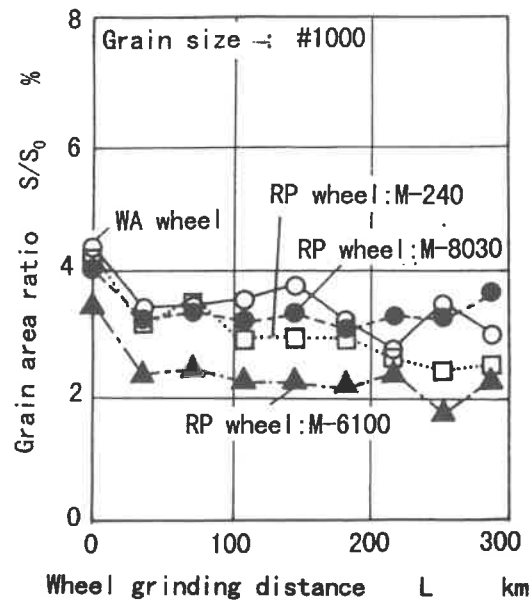


Figure 4 The changes of grain area ratio

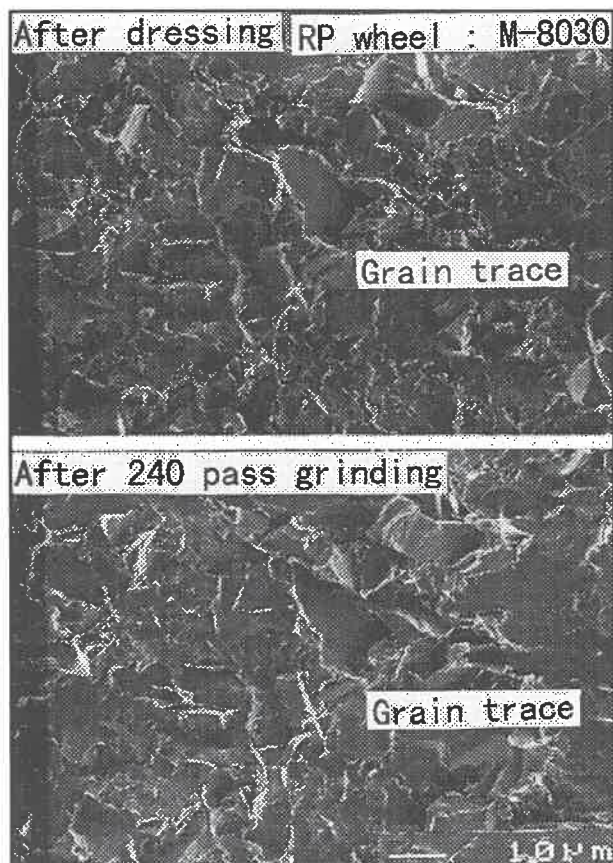


Figure 5 The worn surface of wheel

grain releasing and the contact between workpiece and bond.

However, a lot of sharp abrasive grains still remain

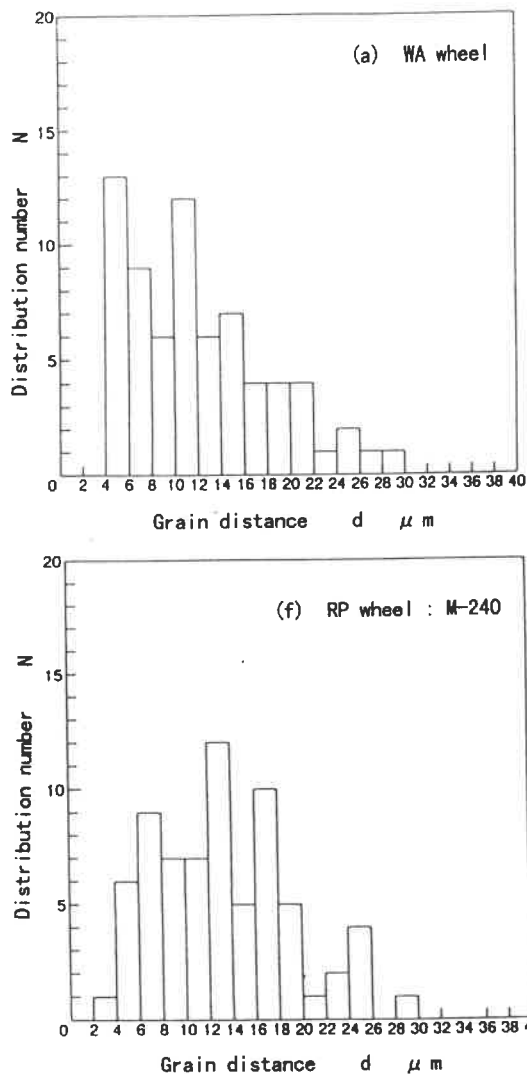


Figure 6 Changes of grain distance distribution

on the worn surface after 240 pass grinding in spite of the abrasion of bond.

Figure 6 shows the changes of grain distance histogram which presents the uniform or random distribution of abrasive grains. If the peak of the numbers is on the short grain distance, the grains do not distribute uniformly and rather scattered with the rough and close parts.

The peak of histogram is presented on the short distance in the case of WA wheel, but that is on rather large distance in the RP wheel case of M-240 resin which has the viscosity of 10-30 cps 25°C. This means that abrasive grains uniformly distribute in the resin. Generally, the low viscosity of resin permits the uniform distribution of grains.

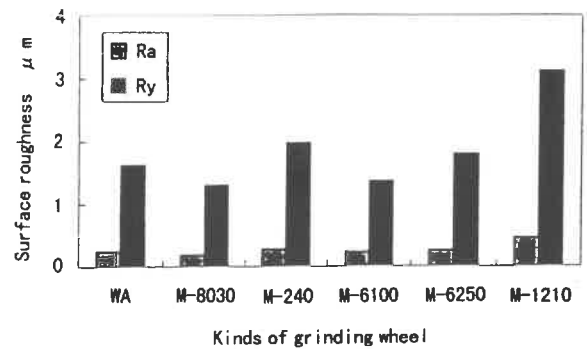


Figure 7 The results of surface roughness

Figure 7 shows the measured results of surface roughness. The ground surface roughness by RP wheel are smaller or equal to that by WA wheel. But, there is no clear relation between the surface roughness and the uniform distribution in such a little high roughness. M-1210 resin has the large viscosity, low tensile strength and large elongation. The abrasive grains may be push into bond by the grinding force. Thus, the large surface roughness was measured on the pre-ground surface without scratching by abrasive grains.

CONCLUSIONS

The creative idea bases on the apply of the rapid prototyping principal and technology which the resin bonded grinding wheels (RP wheel) are composed with the stepped layers of the ultra violet curing resin mixed with abrasive grain.

The following conclusions are obtained by the careful investigations about the mechanical and curing characteristics of resins, the manufacturing method of polishing disk and grinding wheel and the verification of their machining abilities by the polishing and grinding tests.

1. It is cleared that the cured depth and width of the ultra violet cured resin increase with the long exposure time and only 5 to 10 sec exposure of ultra violet is eligible to fabricate a small area of grinding wheel surface.
2. It is possible to make the polishing disk and grinding wheel which hold tightly and distribute uniformly abrasive grains. The grinding ability of RP wheel is large due to the enough bending strength and elastic modulus.

Fine Grinding of Engineering Ceramics as Substitution for Double-Side Lapping

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1 INTRODUCTION

Various components in industrial applications demand high standards on surface roughness, and on dimension and shape accuracy. These are for example ball bearing rings, piston rings and ceramic sealing washers. Due to the specific process kinematics, the manufacturing process of plane lapping meets the high quality demands and therefore presents a common finishing operation. During lapping, an oil- or water-based lapping suspension of dispersed lapping grains consisting of silicon carbide, boron carbide or diamond is supplied to the working gap. On account of a rolling motion of the lapping grains, small material particles break out. This is promoted by a continuous deformation and embrittlement of the workpiece material [1]. As a consequence of the normal load of the lapping wheels, the abrasives are splintered and have to be removed after leaving the contact zone. A considerable amount of lapping suspensions has to be disposed. Besides the high disposal costs, the low removal rates and a high cleaning expenditure are further disadvantages which have led to the development of a new manufacturing process [2]. The process is not clearly defined in the standards but is commonly specified as „fine grinding“, „plane honing“ or „lap-grinding“ [1]. Fine grinding combines the advantages of the lapping kinematics with ecological and economical improvements by the use of bonded superabrasives.

The results of this paper were achieved with support of the German Ministry of Research and Technology (BMBF).

2 FINE GRINDING

Fine grinding presents a new double-side surface grinding process for high precision machining. The

principle is characterised in [figure 1](#). The kinematics are equivalent to lapping with restricted guidance. During fine grinding, the workpieces are guided in a carrier system which is driven by an inner pin ring. The carrier system rolls off on the fixed outer pin ring so that the workpieces move in a cycloid motion between the rotating grinding wheels similar to a planetary gear [1]. Due to these specific kinematics, high dimension and shape accuracy are expected to be achieved. The use of bonded superabrasives leads to a significant reduction of disposal costs and to higher removal rates. The reduced pollution of the workpieces means a lower cleaning expenditure and offers the potential for an automation of the process.

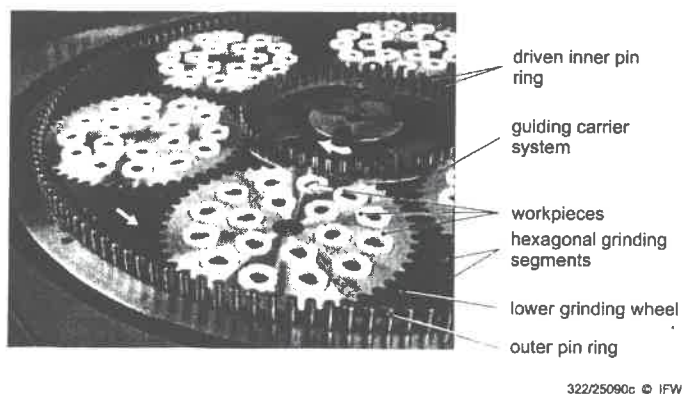


Fig. 1: Principle of fine grinding

In comparison to conventional grinding processes, material removal during fine grinding results from high contact areas between workpiece and grinding tool with comparatively low cutting speeds.

3 FINE GRINDING MACHINES

The fine grinding machines are based on further developments of conventional double-wheel lapping machines operating with a force-controlled feed system and cycloidal workpiece feed, with the grinding wheels being a substitution for lapping wheels. The drive speeds of the two grinding wheels and the inner pin ring are controlled independently of each other and are an important factor in influencing the machining result and the shape of the grinding wheels.

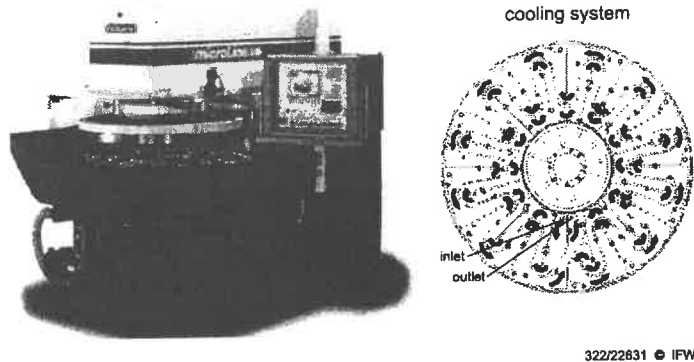


Fig. 2: Peter Wolters double wheel fine grinding machine AC 1200-F microLine[®]

As an example for fine grinding machines, the Peter Wolters fine grinding system AC 1200-F microLine[®] is illustrated in figure 2. The machine is suited for workpieces with a diameter of 10 to 340 mm and a height of 0,4 to 90 mm. The cutting speed in grinding is much higher than in conventional lapping processes. Due to the modification of the gearing, the cutting speed is increased, for example with the fine grinding machine AC 1200-F, a cutting speed of 10 m/s or more can be achieved. As machining forces are higher than in lapping the power of the drives is increased by strong AC servo motors.

Early machine concepts with hydraulic load control were replaced by machines with a pneumatic / mechanical loading mechanism. The fine grinding wheels are loaded by means of a pneumatically-driven pair of bellows which is connected to the upper fine grinding wheel via a rocker. Unlike loading systems which are activated by hydraulics, this pneumatically-driven system is also suitable for machining of thin and sensitive workpieces because of reduced friction as well as more sensitive adjustment and control of the working pressure. Due to the need of high rigidity and to minimise the influence of changing cutting forces during machining, a strong, pre-stressed slide supported by a low-friction bearing is used. Another special feature of the machine concept for grinding wheel diameters of 700 mm and larger is the cooling system in the hub flanges, which leads to a more even temperature distribution and a reduction of distortion of the grinding wheels due to heat (figure 2).

4 FINE GRINDING TOOLS

Diamond is applied because of its high hardness and wear resistance during fine grinding of ceramics, tungsten carbide etc. [3]. Fine grinding wheels are commonly equipped with a resin or vitreous bond system. The application of metallic bonds in fine grinding proves to be disadvantageous in most cases. Because of the high thermal conductivity of the metallic bond process heat leads to different thermal expansions and an unevenness of the working wheels. These disadvantages may be prevented by resin bonds which, in addition, enable the machining of thin workpieces due to lower process forces. Resin bonds are also characterised by their low price and simple truing possibilities. The development of vitreous bonds has even led to further improvements. In detail, the influence of temperature on the workpiece, the tool wear and the grinding forces were reduced. As a result of the lower contact pressure, thin as well as thick workpieces can be machined. Furthermore, fluctuating cutting capacities can be avoided which occurred especially during the machining of steel materials with resinoid bond systems [2].

Besides a complete area abrasive layer with grinding segments a segmented abrasive layer with pellets is also suitable. This design ensures a sufficient removal of heat and cutting products off the contact zone and is therefore favoured in recent times. The use of pellets ensures a high flexibility in wheel design, enabling wheels to be made in different sizes and for a wide variety of machining tasks. For exam-

ple, the pellets may be arranged in a coordinate, a ring-shaped or a spiral design. Fine grinding wheels with a hexagonal layout with grooves for the removal of coolant and chips are also commonly applied. This wheel design is predestined for the machining of workpieces with small diameters. Using large-surface grinding coatings, the concentration of the abrasives is varied from the outer to the inner radius in order to achieve a uniform wear of the grinding tool. Dependent on the present task, diamond grains in FEPA standard grain sizes of 181 to 46 μm are applied. In addition, grinding wheels with micro grains are used to achieve high surface qualities ($R_z < 0,5 \mu\text{m}$).

5 PROCESS RESULTS

In the following, process results shall be discussed which were achieved while machining ceramic sealing washers. The influence of tool and coolant is shown in figure 4. Grinding wheels with a grit size of D20-D30, D91 and D107 in a vitreous and resin bond were applied. 108 components consisting of aluminium oxide with a diameter of 34,4 mm were ground in one charge. A mineral oil and a water base fluid were used as coolant. The

stock allowance was 400 μm . The truing of the grinding wheel is fulfilled with a lapping suspension of silicon and boron carbide. A self-sharpening effect of the grinding tool and a constant process course can be achieved after a short run-in period by a suitable choice of the process parameters. For this reason the figure shows the machining results which were produced in the 10th charge after the initial truing of the grinding wheel.

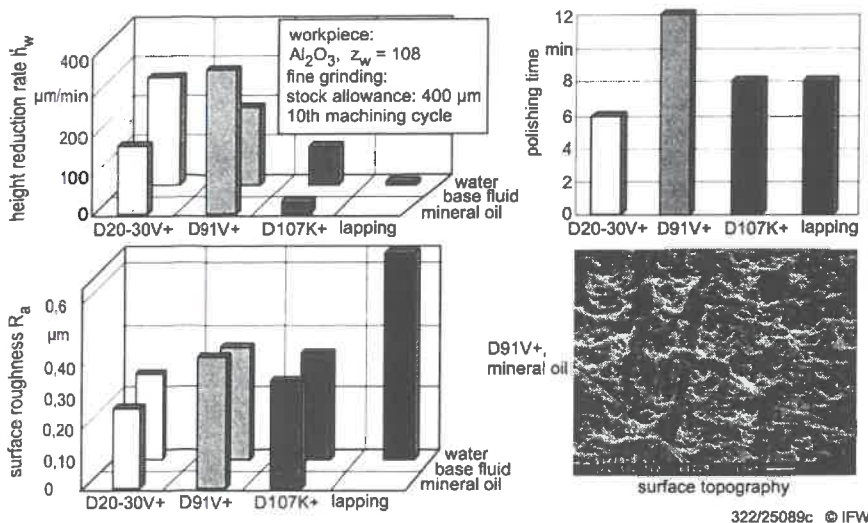


Fig. 4: Influence of tool and coolant

fine grinding is not appointed by the operator but results from the machining conditions. The grinding wheel D91 in a vitreous bond leads to the highest material removal rate of more than 350 $\mu\text{m}/\text{min}$. Significantly smaller material removal rates result from the application of the grinding wheel D107 in a resin bond in spite of the higher grit size. This is caused by the better self-sharpening effect of the vitreous bond system. The influence of the coolant can not be generalised for all tool specifications. Whereas the grinding wheel D91 leads to the highest material removal rate in combination with the mineral oil, the application of the water base fluid is advantageous with the grinding wheels D20-30 and D107.

The best surface roughness R_a of 0,25 μm is produced by the grinding wheel D20-30 because of the smaller grit size. Regarding the influence of the coolant, only slight differences may be noticed in combination with the grinding wheels D20-30 and D107. In contrast, the grinding wheel D91 produces significantly smoother surfaces when applying the water base fluid instead of mineral oil because of the smaller material removal rate. The surfaces of the fine ground workpieces are characterised by undefined crossing grooves referring to ductile as well as to brittle material removal mechanism.

Following the lapping or fine grinding process the ceramic sealing washers have to be polished in order to improve the surface quality. This machining process is cost-intensive due to high personal-expenses so that the polishing time is a predominant factor for the economy of the pre-machining operation. As illustrated in figure 4, the necessary polishing time after lapping amounts to 8 minutes. Whereas the grinding wheel D107 leads to the same result, the polishing expenditure after fine grinding with the tool

D91 is higher in combination with a shorter grinding time. Significant improvements may be achieved by the application of micro grains of a grit size of D20-30. This reveals that to lapping not only the process time of the pre-machining operation but also the polishing time may be considerably decreased by fine grinding.

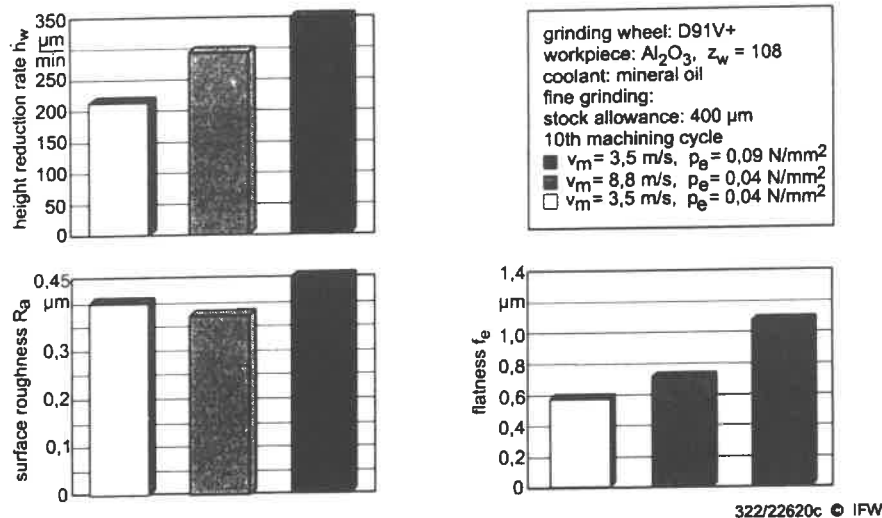


Fig. 5: Influence of cutting speed and working pressure

cutting speed of 3,5 m/s and a working pressure of 0,04 N/mm². As a consequence of higher contact lengths, an increase of the speed of the grinding wheels leads to a higher material removal rate and a better surface quality. Due to increased cutting depths, higher working pressure creates a higher material removal rate and a deterioration of the surface quality [1].

The flatness of the workpieces is also influenced by the machining parameters. A higher medium cutting speed and working pressure leads to a deterioration of the shape accuracy. Due to the higher stock removal rates, disturbances have a higher influence on the process results.

6 CONCLUSIONS

Fine grinding presents a new double-side plane grinding process which is suitable for a great variety of materials and geometries. The process is based on lapping kinematics and cycloidal workpiece movement and is expected to achieve high dimension and shape accuracy and surface qualities. Recent developments of an adapted machining concept and appropriate grinding wheels in combination with extensive research work on process performance have led to a breakthrough of fine grinding in mass production.

Similar to other applications, during the finishing operation of ceramic sealing washers, fine grinding can replace lapping. Despite higher machine and tool costs, the reduction of cost per part can be significant. Through the use of ultrahard diamond abrasives in resin or vitreous bonds lower waste disposal costs, simple workpiece cleaning and higher material removal rates can be achieved. Besides, the machining time in the subsequent polishing process can be reduced by a suitable choice of the grinding conditions. The potential for an automation of the process contains further possibilities to save production costs.

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IN-PROCESS-DRESSING OF FINE DIAMOND WHEELS FOR TOOL GRINDING

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1 ABSTRACT

In tool grinding operations especially the ceramic cutting tool materials two main aims are important - high production efficiency and high precision cutting edges with minimum surface and sub-surface damage. Using conventional grinding conditions these two aims can not be combined without essential restrictions. The in-process-dressing enables an efficient use of fine grained grinding wheels to obtain a superior cutting edge quality.

2 INTRODUCTION

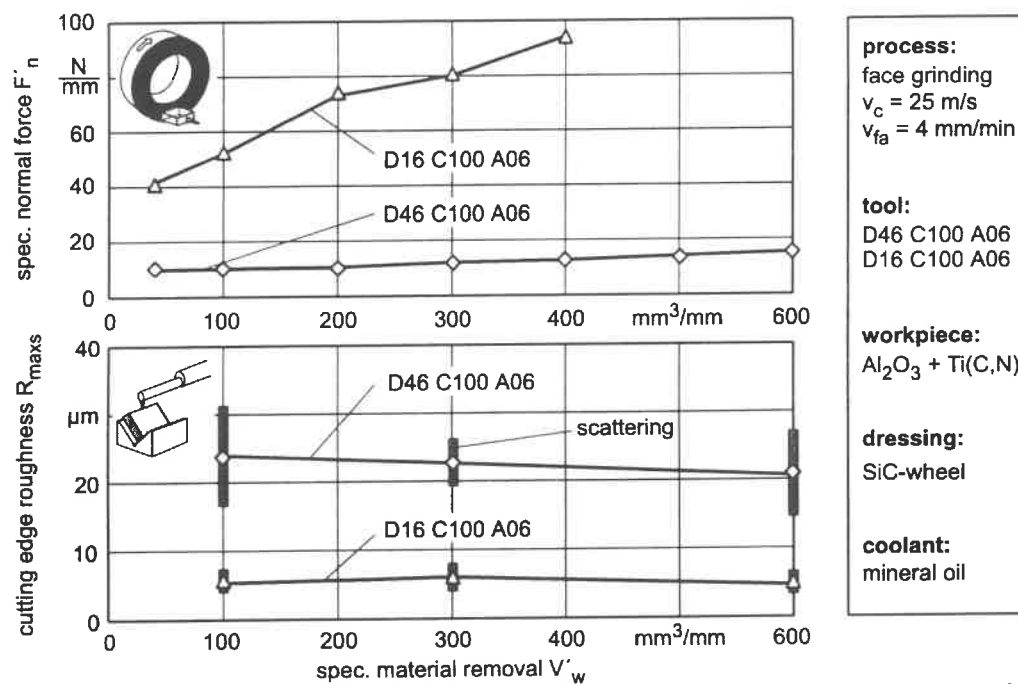
In machining operations like milling or drilling mostly indexable inserts are used. These applications set high demands on the cutting tool material. Besides the dimensional quality in size and shape the advanced mechanical properties of the material as superior hardness and toughness even at high temperatures are most important. Significant for an accurate machining performance is a precise and regular shaped cutting edge. The cutting edge quality is generated during the tool grinding operation.

The main aim in the production of cutting tools is to increase the output and simultaneously to decrease the surface and sub-surface damage of the cutting edges, especially in grinding ceramic cutting tool materials. These materials lead to break-outs and micro cracks at the cutting edges which cause a short life time in the subsequent machining operation. In conventional grinding these two targets - high efficiency and minimum surface damage - are contrary. To avoid these problems, a new technique for sharpening by electro contact discharge dressing (ECDD) is shown and compared to the conventional abrasive sharpening in this paper.

3 CONVENTIONAL GRINDING WITH SMALL GRAIN SIZES

In grinding of Al_2O_3 -ceramics reinforced with $Ti(C,N)$ the cutting edge quality can be easily influenced by using smaller grained grinding wheels. The cutting edge quality is mainly depends on by the mechanical load during grinding. By reducing the grain size in a grinding layer, the number of grains increases by the square of the grain size reduction. For example due to the reduction of the average grain size from $d_G = 41 \mu m$ (D46) to $d_G = 16 \mu m$ (D16) about seven times more grains are available in the grinding layer [1]. The big enhancement of the higher number of active grains means a reduction of the mechanical treatment by each single grain, because the same material is removed by a higher number of active grains [2]. The lower load on the cutting edge leads to better cutting edge qualities (Fig. 1).

The measurement of the cutting edge roughness is based on the measuring and evaluating principle of the maximum peak-to-valley height R_{max} . With an increase of the material removal V'_w the edge roughness decreases slowly. Simultaneously the grinding forces rise when using small grained grinding wheels (Fig. 1). The grinding performance of these grinding wheels is reduced because only a smaller chip volume is available. In addition to that, small grained wheels are very critical against a variation of the grain protrusion. The conventional use of fine grained wheels generates very high grinding forces and unstable processes.



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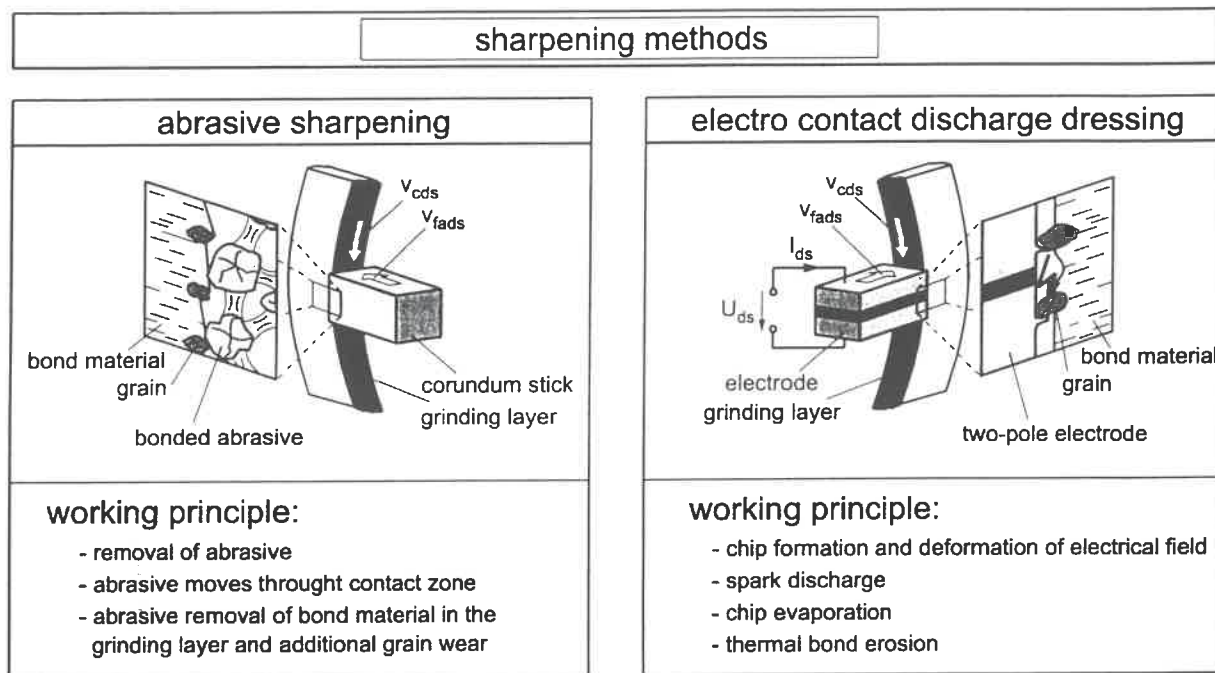
Fig. 1: Grinding forces and cutting edge quality in grinding

Therefore an effective grinding process with small grained grinding wheels for advanced cutting edge qualities is only possible with several restrictions. It is necessary to reduce the influence of the wear mechanisms of the grinding wheel by using more dressing processes and lower material removal rates. These compromises in order to reach high cutting edge qualities make the grinding processes very uneconomic because of the high amount of nonproductive time. One solution to these problems in grinding super-hard and tough materials with small grain sizes is the usage of continuous in-process-sharpening techniques. The main aims of this additional process is to preserve the grinding ability of the grinding layer and to reduce the influence of several wear mechanisms on the grinding process [3].

4 GRINDING WITH IN-PROCESS-SHARPENING

Most important is a constant grain protrusion in the grinding layer especially in grinding with very small grain sizes. Therefore it is necessary to remove all blunt grains and carry out the workpiece material in order to avoid chip loading. In a well-balanced in-process-sharpening the grinding process is characterized by constant grinding forces and steady workpiece qualities.

For in-process-sharpening there are two main working principles (Fig. 2). On one hand the typical abrasive sharpening method with a corundum stick is used. This working principle is characterized by a high additional grain treatment during the sharpening process. That is also a reason for distinct increase of the grinding wheel wear. On the other hand the sharpening by ECDD is characterized by a less grain strain. Therefore the additional grain wear by sharpening is minor. The real sharpening effect arises from the thermal erosion of the bonding material. This process removes the electrical conductive bond material by a local thermal treatment. It is characterized by the voltage between the two electrodes U_{ds} , the current I_{ds} and the electrode material removal rate Q'_{ds} which is given by the geometry and the feed speed v_{fads} .



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Fig. 2: Sharpening methods in grinding

The in-process-sharpening reduces the influence of wear mechanisms. Therefore constant grinding forces and stable processes are possible [3],[2]. The grinding force level depends on the sharpening intensity [5]. In ECDD the intensity is easily adjustable by the sharpening voltage U_{ds0} . The increase of the sharpening voltage generates a higher grain protrusion in the grinding wheel and the grinding forces are reduced. Similar to this, an increase of the abrasive stick removal rate leads to lower grinding forces in grinding with continuous abrasive sharpening.

Fig. 3 shows the comparison between these two sharpening methods concerning the grinding wheel wear Δr_s after a specific material removal of $V'_w = 1000 \text{ mm}^3/\text{mm}$. The three parts of the diagram represent different sharpening conditions with a variation of the sharpening intensity. The comparability of the grinding processes is adjusted by approximately equal grinding forces in grinding with abrasive sharpening and ECDD. The grinding wheel wear Δr_s is shown as a total of the wear depending on the grinding process Δr_{sw} (hatched area) and the additional wear through sharpening Δr_{sds} (white area).

Small sharpening intensities cause low additional wear. When using higher intensities the wear increases, but it is less in sharpening by ECDD. This shows the technological advantage of the electro contact discharge process in sharpening. The grain treatment is very low and therefore the grinding wheel wear is reduced.

5 SUMMARY

The new technique for in-process-sharpening by electro contact discharge enables to adjust a constant grain protrusion and grinding forces independent of the used grain size. By applying ECDD it is possible to use fine grain grinding wheels with the same material removal rates and high material removals as coarse grained wheels. The material treatment and the final cutting edge quality can be increased by using small grain grinding wheels. Therefore the grinding with in-process-sharpening by ECDD enables stable processes by the reduction of grinding forces and increased grinding tool life times.

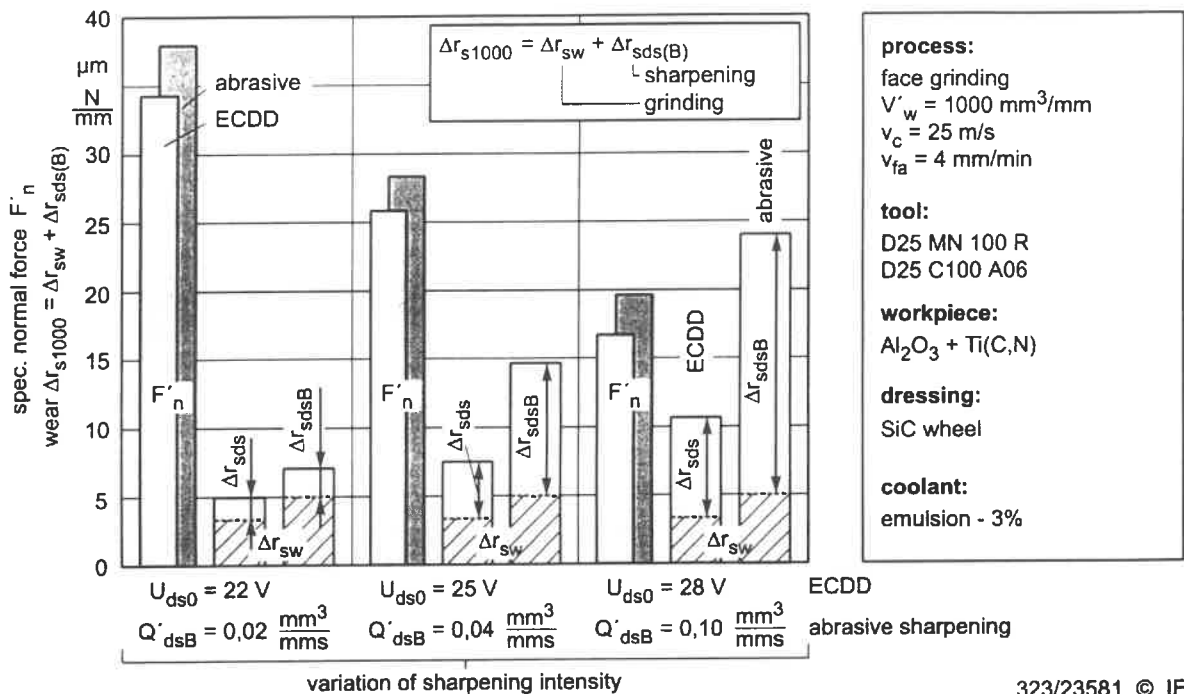


Fig. 3: Comparison between abrasive sharpening and ECDD

ACKNOWLEDGMENTS

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Manufacture of microstructures by ultrasonic lapping

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Introduction

Machining processes with ultrasonically driven tools are expected to be particularly suited for the manufacture of microstructures and microparts due to typically low process forces. Therefore, ultrasonic lapping is applied to generate microscopic geometries in brittle materials, including ceramics, glass and silicon. The microstructures vary from simple boreholes to grooves and pockets with a width of 10 μm to 500 μm and aspect ratios larger than 5:1.

Since the dimensions of lapping tools and grains have been significantly reduced, process parameters for these machining conditions have to be explored. The achievable geometrical tolerances, surface integrity and tool wear were studied as a function of vibration amplitude, feed velocity and grain-size of the lapping medium.

Design of ultrasonic lapping machine

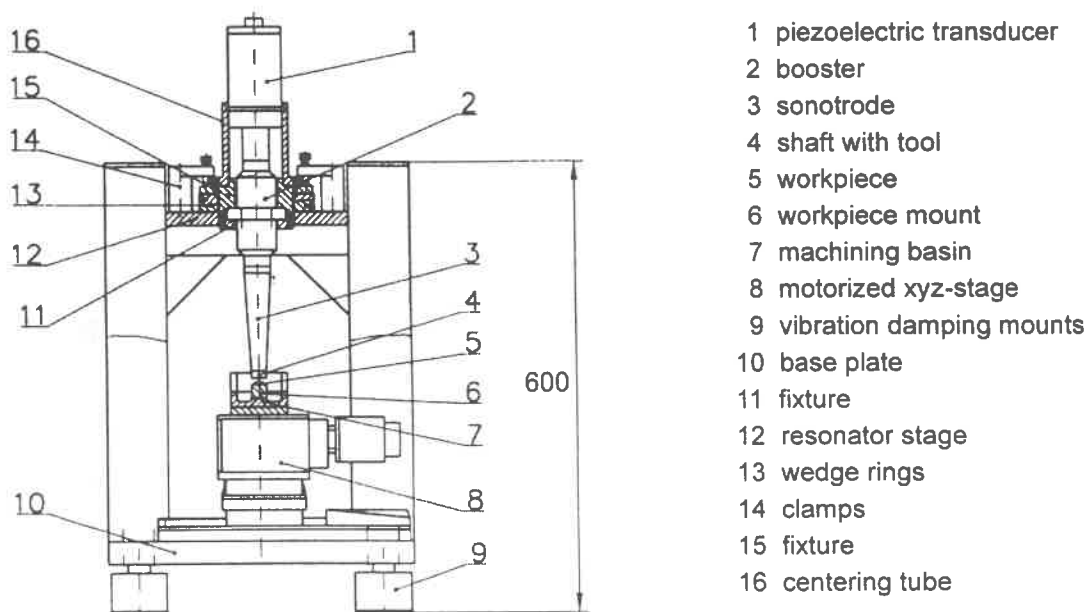


Fig. 1: Design of ultrasonic lapping machine

In ultrasonic lapping hard grains, suspended in a fluid or paste are crushed into the surface of a brittle workpiece by a longitudinally vibrating tool. While the tool is predominantly deformed in a plastic mode, microscopic cracks are induced in the surface of the workpiece at each impact of a - typically blunt - grain [KLO96a]. Integrated over dwell time, material removal is achieved. Hence, the negative face profile of the tool can be transferred into the workpiece.

For the lapping experiments an ultrasonic resonator with a frequency of 20 kHz and a maximum power of 1 kW was mounted vertically above the workpiece (cf. Fig. 1). All infeed and positioning motions were performed by a numerically controlled xyz-stage with a resolution of 0.1 μm . The workpiece is mounted on the xyz-stage in a tub-type machining chamber, allowing for slide travels of approximately 1 inch in each direction. The lapping media is poured over the workpiece from a nozzle and is circulated by a peristaltic pump. As a lapping medium B_4C -grains of mesh-sizes F1200 to F500 (average grain sizes 7.5 μm to 16 μm) are immersed in water.

Precision turning and assembly of microscopic lapping tools

For the generation of microscopic geometries, minituarized lapping tools are required. In conventional ultrasonic machining hardened steel or PKD are used as wear resistant tool materials. In this project, lapping tools were manufactured either by precision turning or by assembly of cylindrical parts. Lapping tools with diameters ranging down to 100 μm were manufactured by precision turning of steel blanks (cf. Fig. 2, left). Alternatively, microscopic lapping tools were assembled by glueing cylindrical parts such as fibres, medical tubes or ground steel cylinders, into a tool shaft (cf. Fig. 2, right).

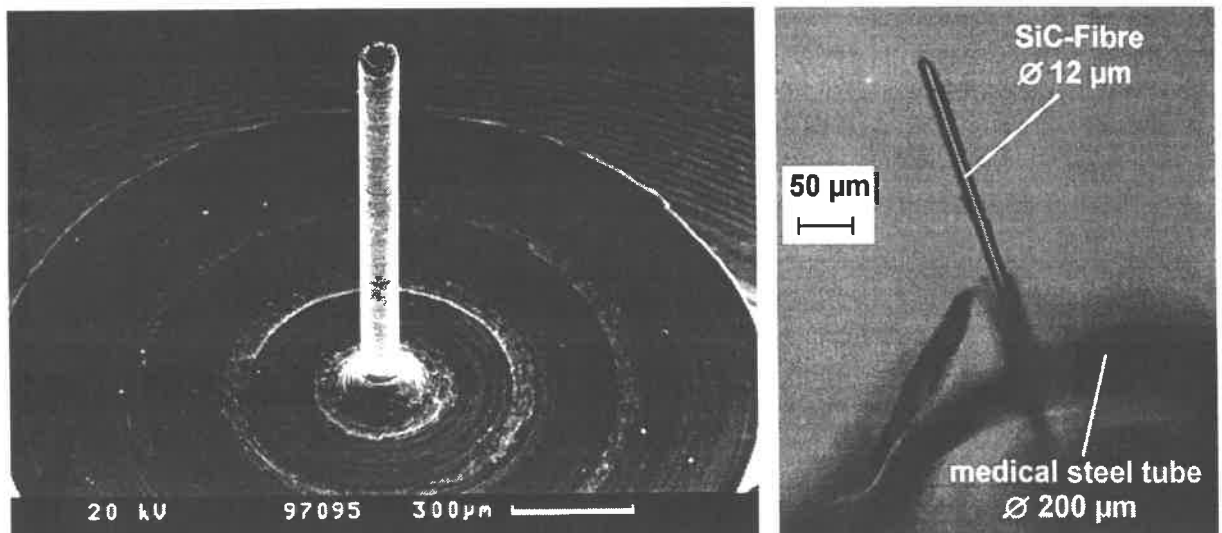


Fig. 2: SEM-image of a precision turned lapping tool, material: steel (left) and micrograph of a tool prototype, featuring a ceramic fibre and a microtube (right).

Manufacture of microstructures

Initially, the generation of microscopic holes was investigated by applying a uniaxial infeed motion normal to the workpiece. Boreholes with diameters ranging from 80 μm to 500 μm were machined in ceramics, glass-ceramics, monocrystalline silicon and BK7 glass. The left part of Fig. 3 shows 360 μm diameter holes, which were machined through a glass plate of 2 mm thickness. The boreholes exhibit a good circularity, but some chipping can be detected at the surface. Due to the brittle material removal mechanism, chipping is a major problem in the generation of microstructures by ultrasonic vibration lapping. Thus, the influence of vibration amplitude and lapping grain size on the visible surface degradation was studied. It was found that chipping can be reduced by using smaller lapping grains, while the vibration amplitude did not effect the amount of surface degradation. Henceforth, lapping media with an average grain size of less than 10 μm (mesh size F1000 or greater) are to be used.

In order to generate microstructures with significant lateral extensions, lateral infeed motions were superimposed onto the normal infeed motion. In this case, the achievable workpiece geometry is no longer confined to the inverted tool geometry. By applying one lateral infeed motion, linear grooves and walls with aspect ratios $> 3:1$ and minimum structure sizes of $100\text{ }\mu\text{m}$ were manufactured in glass and glass-ceramic, as shown in the right part of fig 3.

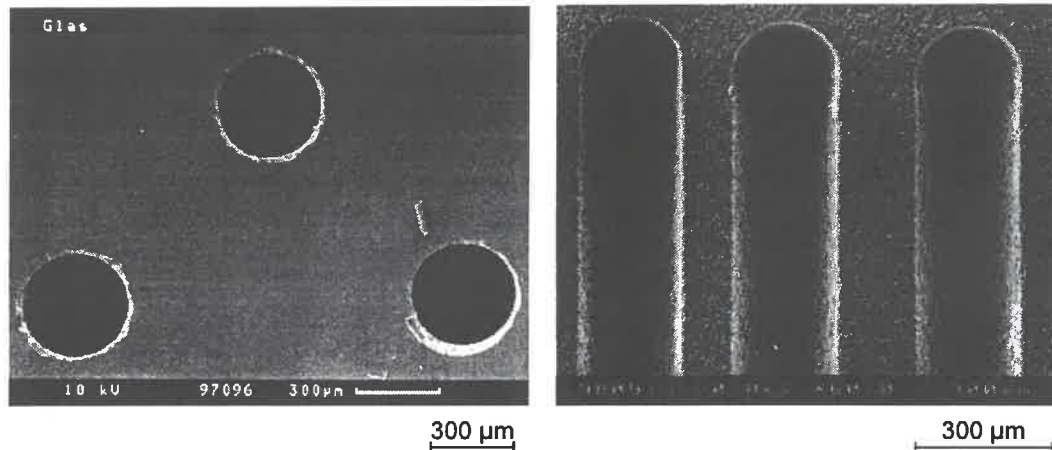


Fig. 3: SEM-image of $d = 360\text{ }\mu\text{m}$ boreholes in glass, depth = 2 mm (left) and grooves in glass ceramic macor, depth = 0.3 mm (right).

Assessment of sub-surface damage

Not only visible radial cracks and chips, but also disruptions normal to the surface have to be minimized by a careful choice of machining parameters. Therefore, the influence of lapping grain size and vibration amplitude on the subsurface integrity in glass, ceramics and silicon is currently being investigated by different etching and polishing methods. The density of defects in machined monocrystalline silicon can be determined by etching with the Yang-base, as summarized by Tönshoff et al. [TÖN90a].

The density and depth of sub-surface damage in machined glass samples is investigated by taper-polishing and etching of very shallow grooves (cf. left part of fig. 4). The polished glass-specimens were etched in HF-acid (40% conc.) for periods of 10 to 30 seconds. The shortest etching times proved to be appropriate for the assessment of subsurface damage in glass, whereas DePiero and Bifano applied the same etching agent for up to 2 minutes in order to assess subsurface damage on ground BK7 surfaces [DEP90]. Apparently, ultrasonically lapped specimens are more susceptible to the etch, which indicates a higher density of defects in comparison to ground surfaces.

After etching, different types of defects can be distinguished in Nomarski micrographs of the surface (cf. right part of fig. 4). The vast majority of defects appear like shallow pits with a depth of only a few microns. However, some exceptionally deep cracks reach down further than 10 microns into the surface, as can be approximately determined by white-light interferometric measurements. The formation of these deep fractures is quite similar to the appearance of median cracks in indentation experiments, according to Lawn and Wilshaw [LAW75].

In the manufacture of microstructures, the latter type of fractures has to be avoided. Currently, the lapping grain sizes and vibration amplitudes are further reduced in order to transfer less energy to the workpiece. Moreover, infeed motions and process strategies have to be further adapted in order to ensure a sufficient supply of lapping grains to the working gap.

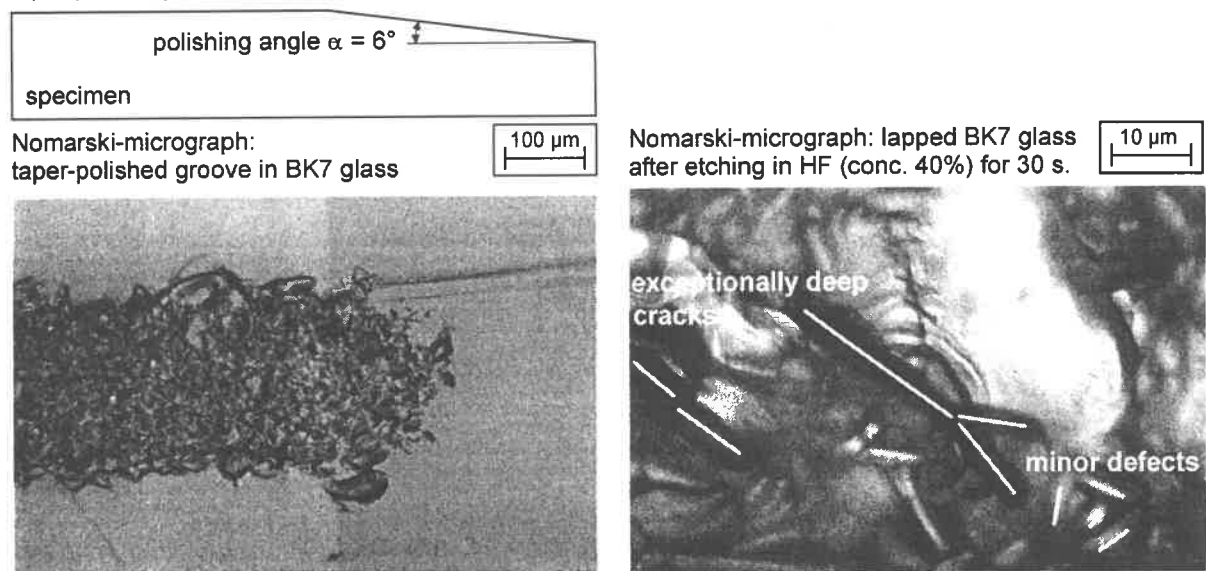


Fig. 4: Micrographs of a taper-polished (left) and etched (right) BK7 glass specimen

Conclusion

Ultrasonic vibration lapping enables the generation of microstructures with high aspect ratios in brittle materials. In contrast to lapping with especially machined one-of-a-kind tools [MAS97], the assembly of micro-tools from commercially available products promises flexible and economic machining operations. Part quality and sub-surface integrity are to be further improved by finding more adapted process parameters. In future stages of this research project, three dimensional microparts will be manufactured by employing five numerically controlled infeed and positioning motions.

Funding of this research project

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A polishing system for aspherical surface generation

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Abstract

A polishing system is designed to obtain an aspherical profile from a spherical one by the form error compensating technique. There are two goals in designing the system hardware. One is to realize the hydrodynamic polishing process in the machining system. The other one is to facilitate the compensating task in creating aspherical surface. A proper hardware and control system is designed accordingly. The experiments indicates that the desired functions can be performed satisfactorily by the designed system.

I. Introduction

A polishing system for shaping an aspherical surface is developed in the ITRI, R. O. C.. This system is aimed to obtain an aspherical surface from a spherical one by the polishing process. It is done by controlling the polishing tool to have a specific dwelling time distribution while the work has a spherical motion. To assure the success of this approach, the polishing system is designed to have near deterministic polishing behaviors. This goal is achieved if the tool can float above work surface in a non-contact or semi-contact manner due to the hydrodynamic effect during the polishing process. [1-7].

Another key of this machining method is to use the form error compensating technique [6 - 9] to eliminate the form difference between aspherical and spherical profiles. It is the goal of the hardware design of system to implement this task easily. The main effort will be on reducing the difficulties encountered in positioning the relative position between polishing pad and work.

II. System Design

The polishing system is composed of a working table, a tool system, a polishing force detector and a controller as shown in Figure 1a.

This table has four degrees of freedom of motions controlled by four different motors. Two are the translational motions and the other two are the rotational ones. Each translational motion is implemented by an individual motor. The directions of translations are orthogonal to each other and forms an X-Y plane. One rotational direction is parallel to the normal of table and the other one rests on the X-Y plane. The first rotational motion is especially designed for an axially symmetric work surface. It is driven by a motor through a belt. To assure a high rotating accuracy, the table is fixed on the granite base.

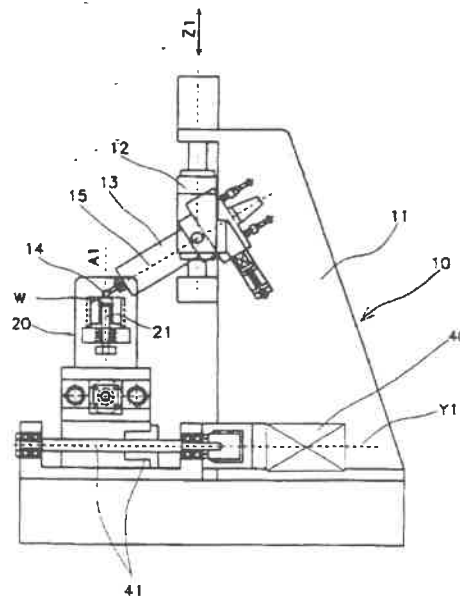


Figure 1a

The second rotation will make the table revolve along an axis with a fixed distance separate between axis and table. It is done by fixing the table base on a crank (as shown in Figure 1b) which is driven by a motor. The main purpose of this rotation motion is to allow the polishing pad to contact the work surface along a spherical trajectory when combined with

the first rotational motion.

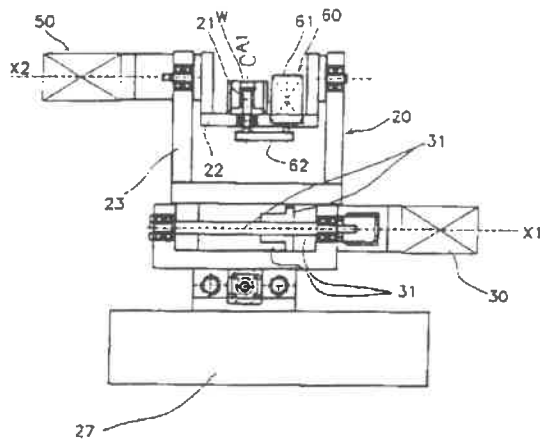
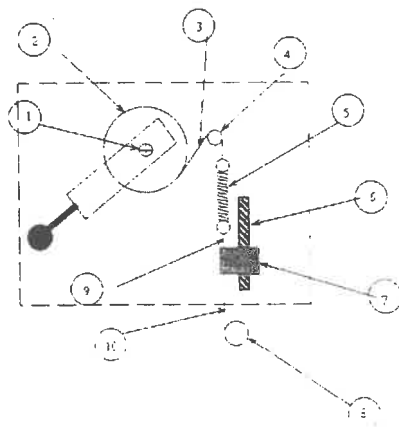


Figure 1b

The tool system has a polishing arm supported on a Z column, which can move up and down by motor Z. In addition to the vertical motion, this arm can also have a rotational movement constrained by a spring mechanism. By proper control of spring deformation and arm's angular movement, the interaction force between polishing pad and work surface can be adjusted. As shown in Figure 2, a string is wired on a cylindrical drum with radius R , which is fixedly attached to and rotates with the polishing arm. The other end of the string is connected to one end of a spring, which again fastens to a load cell by a wire. The load cell is allowed to move on a slider. The motion control of slider can be accomplished by a motor through the action of a wire.



explanation,

1. Polishing arm rotating center O
2. An arc with the center of O, fixed on the polishing arm. There is a inclinometer on the curve.
3. Wire: It is attached at the curve on one end, and at the spring at the other end. The wire is guided by a ball bearing
4. Ball bearing
5. Tension Spring
6. Rail, Slider
7. Load cell, which is fixed on the slider.
8. AC Servo motor rotating shaft: It could rotate CW or CCW to adjust the acting force of the spring.
9. Wire: It is attached at the spring on one end, and at the load cell at the other end.
10. wire: It is attached at the load cell on one end, and at the motor shaft at the other end.

*Inside the region enclosed by dashed line, parts 1,4,6 are fixed at the same base. The relative relation between these three parts are the same.

Figure 2

The force control of polishing pad can be described as follows. Before the pad contacts the work surface, the polishing arm is in a balanced initial configuration. The reading of load cell has a non-zero value which reflects the required force to balance the arm. By moving the arm downward, the pad may start contacting the work. This contact action may first change the angular position of arm and hence the deformation of spring. The change of loading from load cell may indicate the interaction force between pad and work if the friction force of arm, due to angular movement, can be neglected. To maintain the arm in its initial configuration and achieve a desired force at pad, both the slider and arm's vertical positions are adjusted. A multiple-input, multiple-output control system is designed to assured the success of this adjustment. This control system is a rule-based controller.

III. The Polishing Operations

The polishing operations of this system is controlled by a PC controller. The controller consists of five modules. Each module is designed to implement its specific function. These functions are represented by the following modes, which are the JOG mode, Force Control Mode, Working mode, X-Y Home mode, and Configuration mode.

The object of the JOG mode is to supply an environment to allow the operator to manipulate motors independently. Each motor controls its motion based on three commands. They are the Modes, Speed and Direction. The Modes command is to specify the motion of motor. The motion can be the continuous, single step or pulse one. The Speed command is to choose the angular speed of motor. Three levels of speed are designed. The Direction command is to decide whether the motor should run in the clockwise or counterclockwise manner.

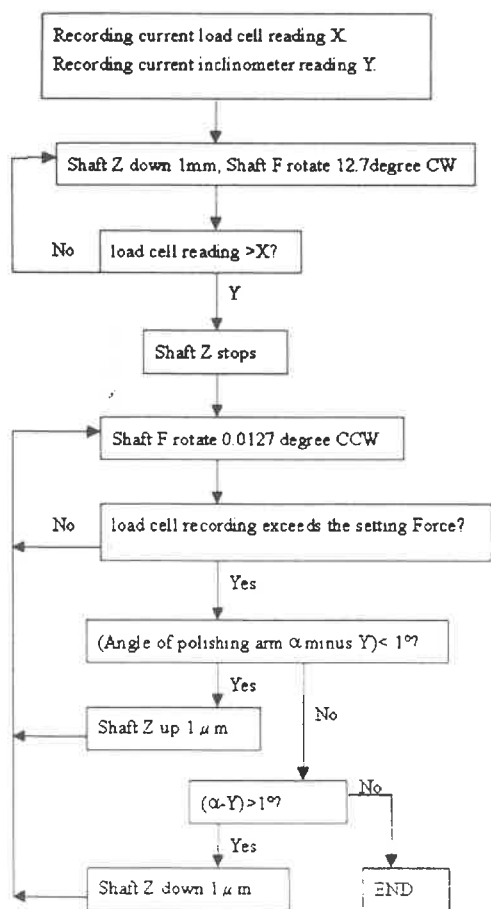


Figure 3

The Force Control mode is to implement the force and angle control tool system. It is a multiple-input, multiple-output controller. Both the number of input and output variables are two. However, in the real implementation, it is simplified into a multi-step single-input, single-output system. The algorithm is shown in Figure 3. At the beginning, the polishing arm is adjusted to a preset initial configuration. Meanwhile, the reading of load cell L is recorded. Then, the polishing pad is moved downward to touch the work by motors Z and F simultaneously. Afterward, the control algorithm comes to the second phase. The object of the second phase is to accurately adjust the force to the desired value. It is done by small adjustment of motor Z and F based on a set of rules. The control procedure is first to meet the requirement of force control for the pad. Then, the arm configuration is adapted. The procedure is repeated until both the force and configuration are controlled to the desired values.

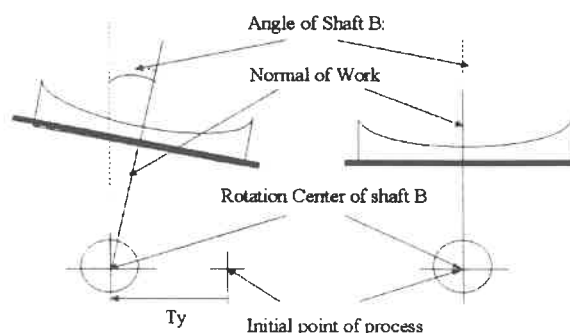


Figure 4

To obtain an aspherical surface from a spherical one by the designed system is done by properly controlling the pad dwelling time while swinging the work along the spherical profile. A schematic representation of this process is shown in Figure 4. The dwelling time is designed based on the method proposed in [8]. It is proportional to the product of the difference between the desired aspherical and spherical profiles and its corresponding radius. In addition to the dwelling time control, the volume removing rate must be kept constant during the polishing process to accurately remove the profile

difference. To do so, the angle between work surface normal and polishing arm is kept unchanged. It is achieved if the polishing arm configuration is maintained and the surface normal at the contact spot between pad and work is not changed. The latter one can be approximately fulfilled if the work keeps swinging along a specific spherical profile during the polishing process. This task is accomplished by the system hardware as described above.

For the demonstration purpose, an example of a process plan file is shown in Figure 5. It can be edited easily by any ASCII editor. There are three parameters in this plan. They are the Y position, B position, and time T. The numbers after these parameters are their corresponding values. For instance, the first command is Y 0 B 0 T 10. This indicates that motors Y and B will stay in the corresponding position (0, 0) for 10 units of time. The second command Y 3 B 6 T 30 asks motor Y staying at 3 and B staying at 6 for 30 units of time.

```

      ⋮
Y 0  B 0  T 10
Y 3  B 6  T 30
      ⋮

```

Figure 5

IV. Conclusion

A polishing system for producing an aspherical surface from spherical one is described. It is accomplished by applying the form error compensating technique to eliminate the form error between aspherical and spherical profiles. The system will realize the hydrodynamic polishing process to implement the task of form error compensation. One feature of the system is to use a simple mechanism to allow the pad to polish the whole surface while maintaining a given geometric relationship between pad and surface normal of work. The maintenance of this geometric relation is important to assure the stability of volume removing rate during the polishing process. The experiments showed that the desired functions of the system perform satisfactorily.

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New approach to smoothing of diamond

- Tribochemical polishing of diamond with some soft powders -

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1. Introduction

For long years (more than 500?) diamond for jewelry or latest diamond tool for diamond turning have been polished with the combination of cast iron tool (so called lap or scaife) and diamond powders. That is one of the state of the art and skill-oriented process. However it is also true that the process have not been investigated from the scientific point of view. We tried to investigate the effect of the cast iron scaife and proposed that hardened steel also can be used as the scaife.¹⁾

This time we focused on the powders. Basically the smoothing of the diamond is done by the mechanical reaction between bulk diamond and powder diamond ;abrasion of diamond by diamond. However we have had some idea that the diamond can be machined by other material powders if some chemical reaction is expected. The chemical reaction should be conducted under the friction among the diamond, the powder and the scaife.

We have tried to investigate this idea experimentally. We have evaluated the effect of eleven kinds of powders which we use frequently in our research for fine polishing. Another purpose of this work is searching possible way to reduce the cost of polishing of diamond. As the diamond powder is so expensive that you can expect cost reduction if the diamond can be machined with cheap powders.

2. Experimental procedure

2.1. Apparatus

Fig.1 shows the conceptual figure of our apparatus. In our experiments we used final polishing (smoothing) process for the diamond tool so as to evaluate the machinability of each powder.

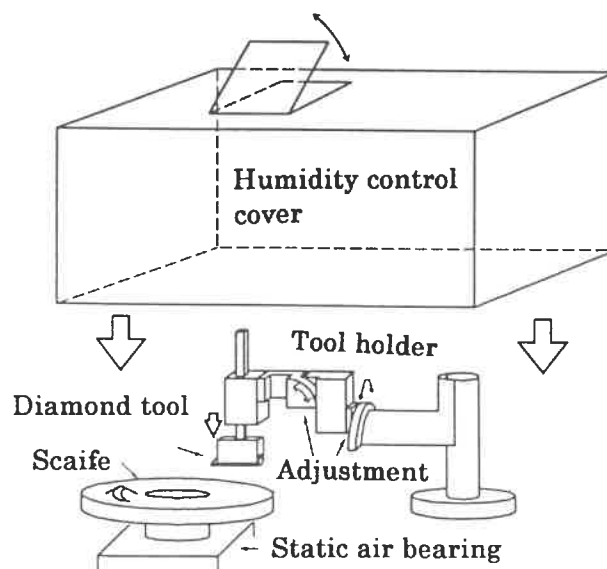


Fig.1 Experimental apparatus

The apparatus is made of the scaife (lap), its driving equipment, and tool holder. We used dead weight method to maintain the polishing pressure constant. The scaife is supported with a static air bearing. We adjusted vertical deviation of the rotating scaife to less than 0.5 μ m. The tool holder

includes an attitude compensation mechanism which makes the whole surface of the tool contact the scaife constantly.

Machining amount of the diamond tools are so small that we measured them by the weight change between before and after the polishing experiment by means of a micro-gram balance (sensitivity 2 μ g). Table 1 shows our experimental conditions. The conditions are same as our finishing (edge sharpening) process. Disc of 160mm diameter made of hardened tool steel was used as the scale.

2.2. Humidity control

As we concentrated to investigate the effect of the surrounding air humidity to the machinability, we put a cover for the scaife and the tool holder.

2.3. Powders

Experiments have been conducted as typical polishing technique using the scaife. The only difference is the use of soft powders instead of diamond powders. Table 2 shows a list of abrasive powders which was used in our experiments. An order of the powders at Table 2 is almost along the hardness of each material (soft to hard). WA and GC are typical and popular abrasive powders; WA is a kind of aluminum oxide and GC is a kind of silicon carbide.

Table 1 Experimental conditions

Workpiece	Natural monocrystalline diamond tool rake surface ; (110)plane
Polishing direction	<001> easily polished direction
Scaife	Hardened tool steel ; ϕ 160mm
Polishing pressure	500kPa
Polishing speed	800m/min (scaife rotation:1800rpm)
Machining time	10 min

Table 2 List of abrasives

No.	name		size(μ m)
1	Titanium dioxide	TiO ₂	1.0
2	Ceria	CeO ₂	0.9
3	Ferric oxide	Fe ₂ O ₃	1.3
4	Ferrosoferric oxide	Fe ₃ O ₄	1.5
5	Chromic oxide	Cr ₂ O ₃	1.4
6	Zirconia	ZrO ₂	0.5
7	Silica	SiO ₂	0.05
8	White alundun	WA	1.2
9	γ -Aluminum oxide	γ -Al ₂ O ₃	0.06
10	Silicon carbide	SiC	1.0
11	Green carborundun	GC	1.2

3. Experimental results and discussions

3.1. Machining with olive oil

At first we machined the diamond with olive oil. In the usual process with diamond powders, the powders are mixed with olive oil to be paste and are spread on the scaife. So each powder replaced the diamond powder in the same procedure.

Fig.2 shows the result of the experiment. In some powders the material removal rate is smaller than that machined without any powder which is pure friction between diamond and steel disc. Generally the material removal rate in this experiments is rather poor in comparison with about 1.6mg per hour of that machined with the diamond powders. The chemical reaction is not so activated when those are pasted with the olive oil.

3.2. Machining with oleic sodium under control of humidity

In the next experiments, we used oleic sodium ($C_{17}H_{33}COONa$) to replace the olive oil. In this case the surface of the scaife is not so wet as the state with the olive oil. During the experiments we realized that humidity of the surrounding air affected significantly the process. So we have controlled and maintained the humidity to 20, 40 and 60 relative percent and tried to clarify the effect of the humidity.

Fig.3 shows the results. The order of horizontal axis comes from a size (diameter) of powders. In all the cases except ceria, the material removal rate in polishing of diamond under 40% humidity is much higher than that under 20% or 60%. That indicates appropriate humidity should be selected in smoothing diamond tool with these powders. The case of using aluminum oxide showed the most severe influence. The material removal rate under 40% humidity was about three times more than that under 20% or 60%. The size of powders seems to make no effects.

Fig.4 shows the same data of Fig.3 along the order of hardness of the abrasives. It also shows the hardness makes no effect to the material removal. The titanium dioxide (TiO_2), γ -aluminum oxide and zirconia (ZrO_2) have shown much larger material removal rate than others. These three shows about 20% rate of that using diamond powders. These three thought to be good candidate to replace diamond powders in polishing of bulk diamond.

4. Conclusion

The diamond can be machined with soft powders. Some tribochemical action is thought to be generated. The effect of the size of powders or its hardness may not affect the material removal rate. It also indicates the exist of the chemical action.

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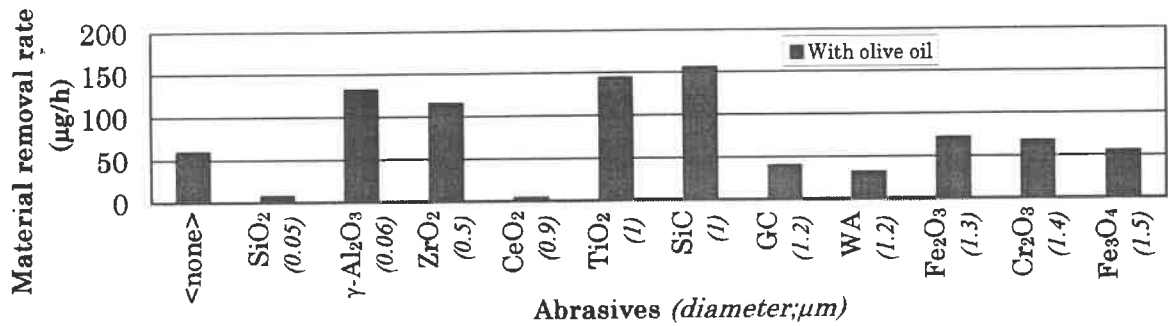


Fig.2 Material removal rate in each abrasive (with olive oil)

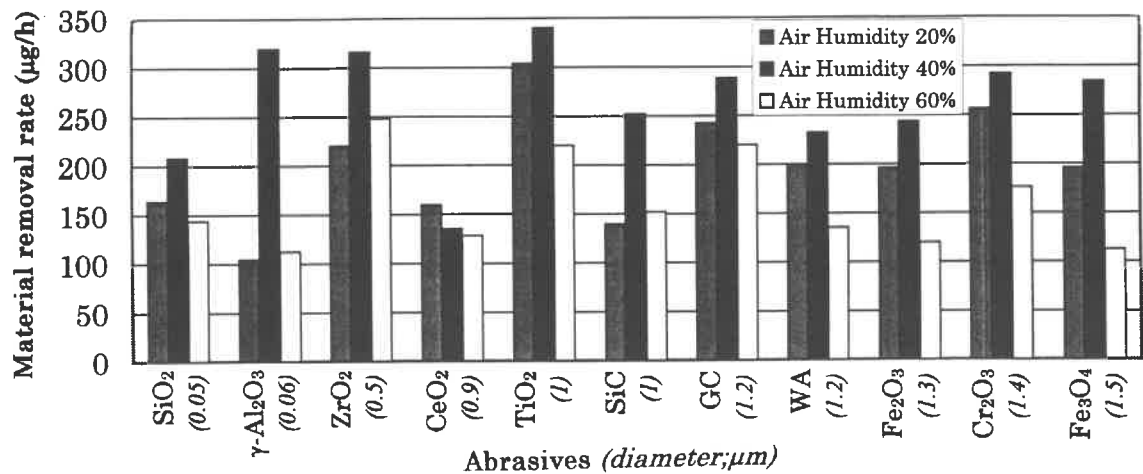


Fig.3 Material removal rate in each abrasive (with oleic sodium)

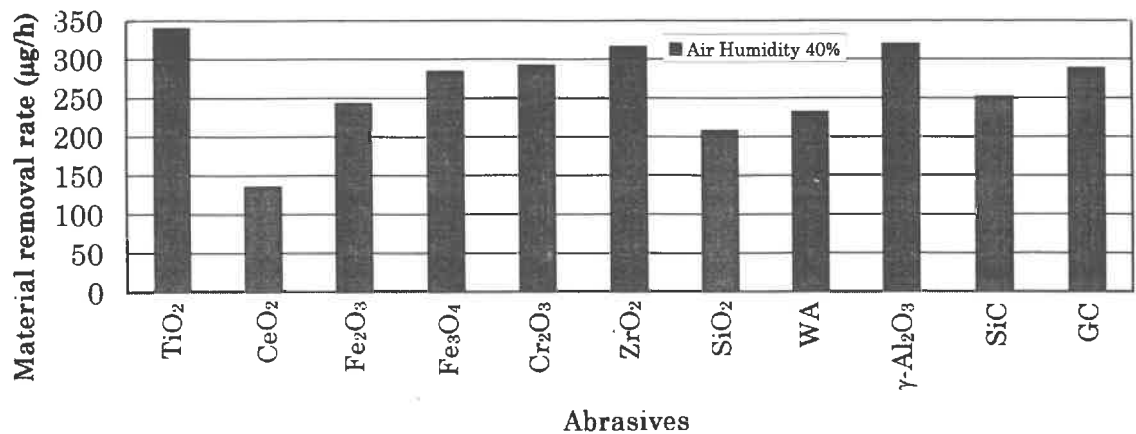


Fig.4 Material removal rate in each abrasive (with oleic sodium)

Influence of the Abrasive Content and Grain Size on Ultra Smoothness Polishing of Glass Magnetic Disk Substrate

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1. INTRODUCTION

To produce the compact magnetic disk of high storage capacity by low cost, the high productive ultra smoothness polishing technic of the magnetic disk substrate has been strongly requested. In previous researches^{1) - 6)}, we have examined the polishing characteristics of electroless nickel phosphorus plated aluminum substrate with alumina abrasives and colloidal silica abrasives, and, on the basis of the result, discussed the optimum ultra smoothness polishing conditions of aluminum substrate. Then the polishing characteristics of the crystallized glass magnetic disk substrate have been discussed by examining experimentally the influence of polishing pressure on the removal rate and surface smoothness⁷⁾.

This is one of a series of researches on finding the optimum ultra smoothness polishing conditions of the crystallized glass magnetic disk substrate.

The influence of the content and size of abrasive grain of polishing fluid on the removal rate and surface smoothness in the polishing of the glass substrate with cerium oxide abrasives is investigated.

2. Experiments

The schematic view of the polishing machine⁷⁾ used is shown in Fig.1. The main experimental conditions are listed in Table 1. The crystallized glass substrate is 2.5 inch type of 65mm in outer diameter and 1.1mm in thickness. The substrate surface smoothness formed by lapping before polishing is about $9\mu\text{m}$ (P-V) and $5\mu\text{m}$ (P-V) for 4mm and $240\mu\text{m}$ in measuring length with ordinal stylus profilometer, respectively. The substrate smoothness after polishing is examined from the standpoint of the slight undulation H_w and

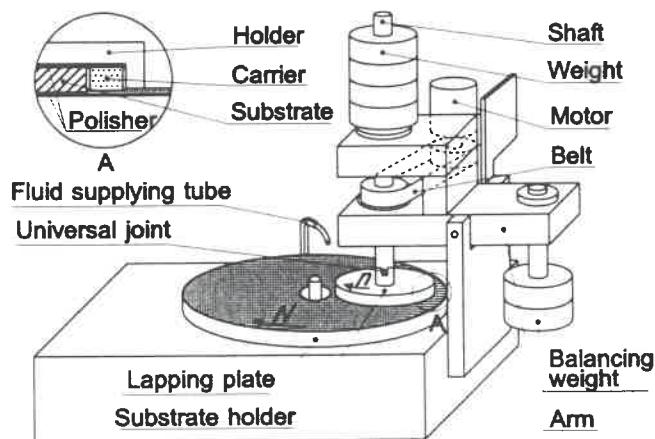


Fig.1 Schematic view of polishing machine

Table 1 Experimental conditions

Substrate	Crystallized glass ($\phi 65\text{ mm} \times \phi 20\text{ mm} \times 1.1\text{ mm}$)
Polisher	Non-woven urethanform type (Hardness: 80°)
Polishing fluid	Abrasives: CeO_2 ($\phi 0.5\text{--}2.3\text{ }\mu\text{m}$) Abrasive Content C_g : 1.0-40.0wt% Supply flux S_g : 6.0 g/min
Polishing condition	Pressure P : 8.1 kPa Speed V_a : 1.5 m/s

the surface roughness H , which are obtained by measuring 4mm and 240 μ m in length with WYKO, respectively. The polishing experiments are carried out in the environment of the class 300 in cleanliness. The removal depth is calculated from the stock removal.

3. Results and discussion

3.1 Influence of the abrasive content

Figure 2 shows the relationship between the removal rate K_t and the abrasive content C_g which ranges 1 to 40%. The K_t denotes the removal depth of substrate per unit time. From the figure, it is found that the K_t increases to the abrasive content of about $C_g=5.0\text{wt}\%$. At more than $C_g=5.0\text{wt}\%$, the removal rate keeps almost the same value. Accordingly the excessive supply of abrasives is considered to be useless for the improvement of productivity in the polishing of the glass substrate.

The change of the surface smoothness with the removal depth for five kinds of abrasive contents is shown in Fig. 3. It is obvious from the figure that both the slight undulation H_w and the surface roughness H decrease rapidly in the initial polishing process for all kinds of abrasive contents. After the initial process, both the H_w and the H decrease gradually as polishing process proceeds and finally converge to the terminal slight undulation $(H_w)_t$ and the terminal surface roughness $(H)_t$, respectively. The change of the surface smoothness with the removal depth is almost the same in quantitative aspect for all kinds of abrasive contents and therefore is considered to be influenced little by the abrasive content.

The minimum required removal depths being necessary to attain the terminal slight undulation and the surface roughness which are expressed by the sign of $(Tw)_t$ and $(Tr)_t$ are about 15 μ m in $(Tw)_t$ and 10 μ m in $(Tr)_t$ for all kinds of abrasive contents, respectively. The minimum required removal depth depends on the surface smoothness before polishing. Accordingly the surface smoothness formed by lapping before polishing is one of

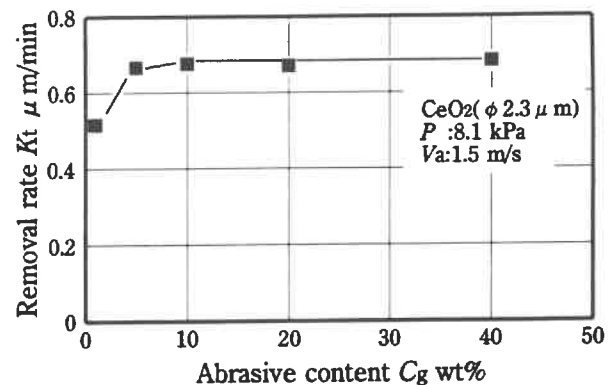


Fig.2 Influence of the abrasive content for the removal rate

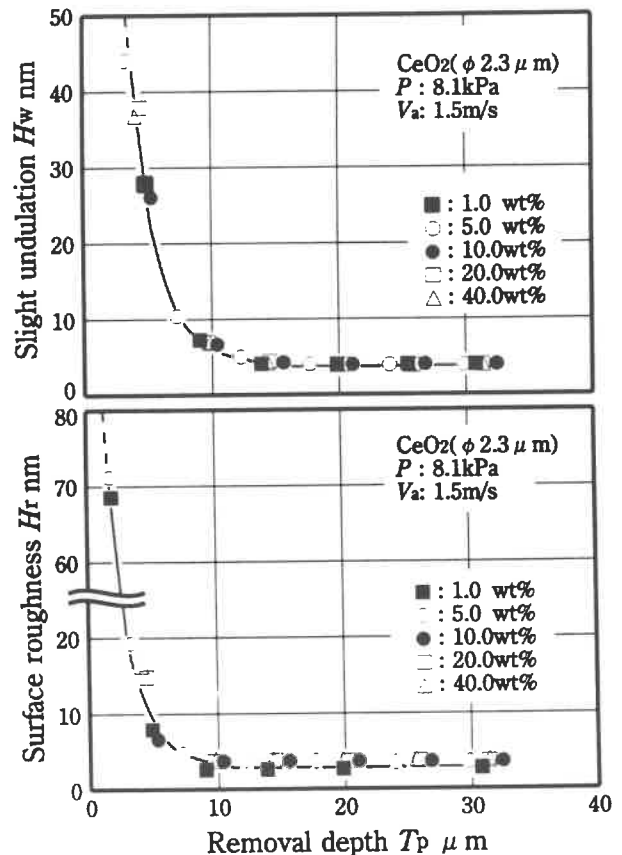


Fig.3 Change of the surface smoothness with the removal depth for five kinds of abrasive contents

the important points related to the productive efficiency in the polishing of glass substrate.

Figure 4 shows the relationship between the terminal smoothness and the abrasive content. From the figure, the $(H_w)_t$ is found to keep about 4nm (P-V) in the whole region of the abrasive content of 1 to 40%. On the other hand, the $(H)_t$ is about 2.5nm (P-V) at 1wt% and increases steeply to about 4nm (P-V) for the increase to $C_g=5\text{wt\%}$. At more than about 5wt%, the $(H)_t$ increases slightly with the increase of abrasive content. Accordingly getting the better surface roughness in the polishing of glass substrate is considered necessary to use the polishing fluid of low abrasive content though the productivity decreases.

3.2 Influence of grain size

Figure 5 shows the relationship between the removal rate and the grain size which ranges 0.5 to 2.3 μm . It is found from the figure that the K_t becomes larger for larger grain size. The rate of increase of the K_t is comparatively steep to the grain size of 1.3 μm . At more than 1.3 μm , the rate of increase becomes slight.

The change of the surface smoothness with the removal depth for four kinds of grain sizes is shown in **Fig. 6**. It is obvious from the figure that both the H_w and the H decrease rapidly in the initial polishing process in the case of all the grain sizes. After the initial process, both the H_w and the H decrease gradually as polishing process proceeds and finally converge to the $(H_w)_t$ and the $(H)_t$, respectively. The change of the surface smoothness with the removal depth is almost the same for all kinds of grain sizes and therefore is considered to be influenced little by grain sizes as well as abrasive content. Thus in the case of all kinds of grain sizes, the $(Tw)_t$ and the $(Tr)_t$ are

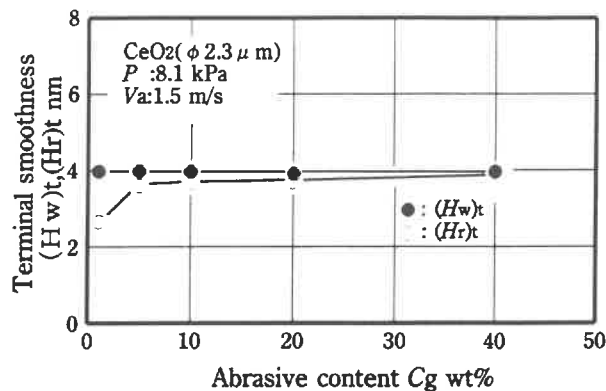


Fig.4 Relationship between the terminal

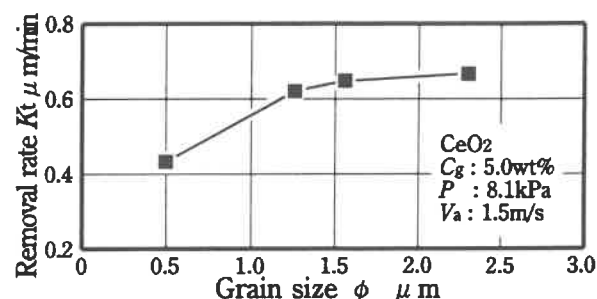


Fig.5 Influence of the grain size on the removal rate

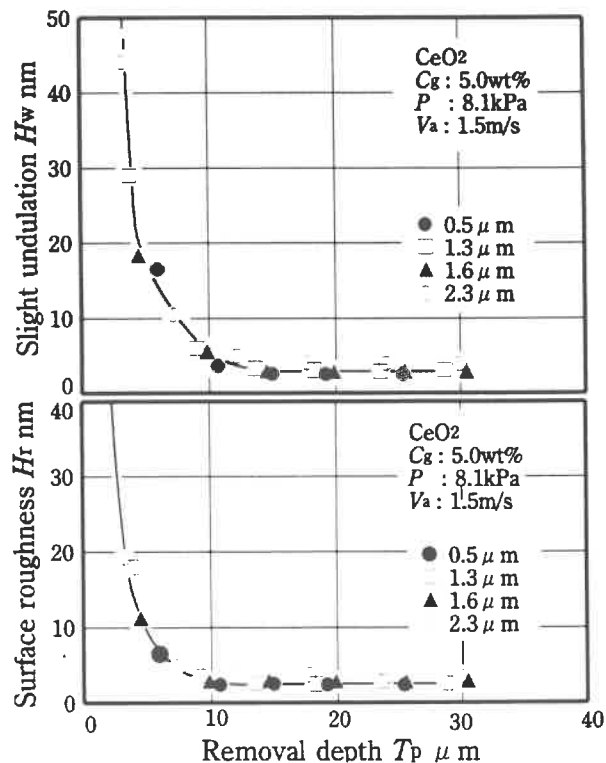


Fig.6 Change of the surface smoothness with the removal depth for four kinds of grain sizes

almost the same removal depth of $15\text{ }\mu\text{m}$ and $10\text{ }\mu\text{m}$, respectively.

The relationship between the terminal smoothness and the grain size is shown in Fig.7. From the figure, both the $(H_w)_t$ and the $(H_r)_t$ are found to increase in proportion to the grain size. Both the $(H_w)_t$ and the $(H_r)_t$ at the grain size of $\phi=0.5\text{ }\mu\text{m}$ are about 2.5nm(P-V) . On the other hand, both the $(H_w)_t$ and the $(H_r)_t$ at the grain size of $\phi=2.3\text{ }\mu\text{m}$ are near 4nm(P-V) . Thus the small grain size is considered better for producing the ultra smoothness glass substrate in polishing.

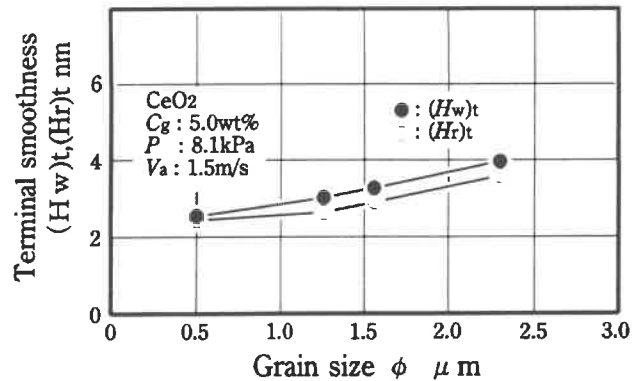


Fig.7 Influence of the grain size on the terminal smoothness

4. Summary

The influence of the abrasive content and grain size on the removal rate and surface smoothness in the polishing of crystallized glass magnetic disk substrate with cerium oxide abrasives is investigated.

The removal rate increases to below a certain abrasive content and a certain grain size. At more than those, the abrasive content and the grain size have little influence on the removal rate. The surface smoothness improves rapidly in the initial polishing process. After the initial process, the rate of its improvement becomes gradually and finally stops at the terminal smoothness. The terminal smoothness deteriorates steeply to below a certain abrasive content and gradually at more than the content. The terminal smoothness is better for smaller grain size.

The best terminal smoothness in the research is about 2.5nm(P-V) in both the slight undulation and the surface roughness at the grain size of $\phi=0.5\text{ }\mu\text{m}$ and the abrasive content of $C_g=5.0\text{wt}\%$.

Acknowledgements

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Dry Polishing of Si Wafer on BaCO₃ Abrasive Disk and its Machining Mechanism

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1 Introduction

In current polishing process in Si wafer production, great amount of chemical slurry of KOH solution mixed with SiO₂ fine powder or colloidal-silica is used and wasted. Not only from the economical consideration in slurry usage and its disposal, but also from the ecological point of view, such chemical slurry should be diminished or eliminated if possible. One of the most innovatory solutions will be the introduction of dry polishing process instead of the current wet polishing. The authors have found that dry polishing of Si wafers can be effectively and stably operated by using fine BaCO₃ powder which is mechanically softer than Si and is able to supersmoothly polish it with mechanochemical effect, remaining no mechanical damage on the Si surface¹⁾. It has already been reported that this dry polishing can be realized not only by supplying loose BaCO₃ fine powder on conventional polishing pads but also by rubbing with bonded BaCO₃ pellets²⁾. The usage of bonded abrasive tool would be more desirable in the practical polishing process in the next century. This paper describes experimental results of Si wafer polishing on a bonded BaCO₃ disk and discusses the polishing mechanism through various physical and chemical analyses.

2 Background of mechanochemical polishing of Si wafers using BaCO₃ powder

BaCO₃ powder has been found to be the most suitable abrasive for mechanochemical polishing of Si wafers¹⁾. It is known that BaCO₃ having the Mohs hardness of 3.0~3.7 is mechanically much softer than Si of which Mohs hardness is about 7 and chemically reactive with it. At the real contact points between BaCO₃ powder and the Si wafer surface, solid state reaction would occur and the very small interfacial reaction products formed at the contact points would be rubbed off during the sliding operation. Neither indentation nor scratching action of the powder would occur at the real contact area because of mechanical softness of the BaCO₃ abrasive and therefore no mechanical damage could be induced on the worked Si surfaces. As the mechanochemical action affects much higher on dry condition than on wet condition, higher polishing rate could be obtained on dry condition without any liquid supply.

On the other hand, polishing by bonded abrasive tool should be more promoted from the view-point of environmental cleanliness and process automation. The previous paper²⁾ made it clear that mechanochemical polishing of Si wafers by a polishing wheel made of bonded BaCO₃ pellets can be effectively operated in grinding-like polishing process even on dry condition. In this paper the possibility of dry polishing of Si wafers performed on a bonded BaCO₃ disk will be investigated in conventional pressure-control type polishing.

3 Polishing experiments and the results

3.1 Forming of a BaCO₃ disk

In mechanochemical polishing the abrasive should act as free abrasive not as fixed one. Even in the case of bonded BaCO₃ disk, the powder should be easily rubbed off from the disk surface and act as loose abrasives. The situation is quite different from grinding operation by hard abrasives. So, 95 wt.% BaCO₃ powder of the mean size of 0.4 μ m and the purity of 99.9% was mixed with 5 wt.% phenol resin bond, pressed at 30 MPa in a forming mold and heat-hardened to a large-size disk having the outer diameter of 380 mm and the inner diameter of 140 mm.

3.2 Polishing procedure

The BaCO_3 disk formed above was set on a rotary-table type polishing machine (EJW—400IFN of Engis Japan). The disk surface was flat faced on machine to the flatness within $\pm 1 \mu\text{m}$. A Si wafer of the diameter of 100 mm which was pasted on a ceramic holder was pressed onto the BaCO_3 disk and polished on dry condition as seen in Fig.1. Figure 2 shows a snapshot of the actual operation. Prior to polishing operation the wafer surface was lapped by $3 \mu\text{m}$ diamond slurry upto the surface roughness R_a about 20 nm. The wafer holder was forced to oscillate with the amplitude of 0, 8 and 20 mm in the radial direction ab as illustrated in Fig.1. As the radial length of the disk is 120 mm, the wafer to be polished overhangs 10 mm both at the disk-outside and -inside when the amplitude is 40 mm, whereas no overhang occurs when the amplitude is smaller than 20 mm. Polishing pressure and disk revolution were fixed to 10 kPa and 40 rpm respectively.

3.3 Machining results

When no oscillation was given, it was found that the central part of the disk contacting with the wafer wore down to $3 \mu\text{m}$ concave shape after 5 minutes polishing and further to $10 \mu\text{m}$ after another 5 minutes operation. The whole surface of the wafer could not be polished within this polishing time. Table 1 shows polishing results obtained when the oscillation amplitude was 8 and 20 mm respectively. It is clearly recognized that the total flatness of the wafer polished 20 minutes with the amplitude of 20 mm is kept in excellent value under $0.3 \mu\text{m}$. The surface roughness also reached the same fine values as those obtained by the primary polishing of the conventional process. An example of the total wafer flatness interferometrically measured after 10 minutes polishing with the amplitude of 20 mm is illustrated in Fig.3.

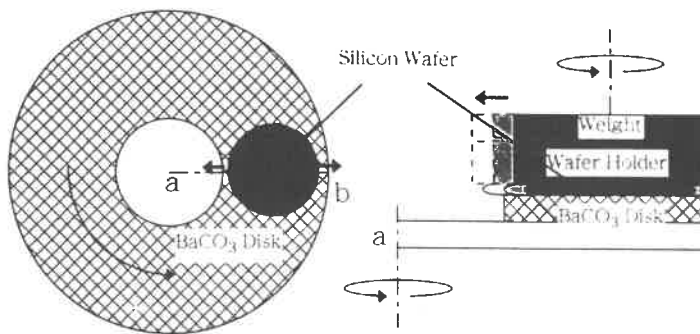


Fig.1 Schematic diagram of the dry polishing

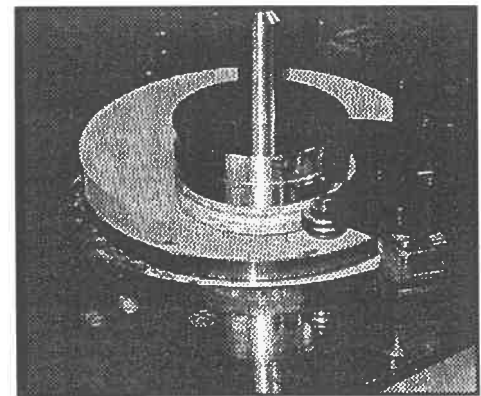


Fig.2 Actual polishing operation

Table 1 Surface characteristics of 100 mm wafer polished by BaCO_3 disk

Amplitude of oscillation (mm)	Polishing time (minutes)	Total flatness (μm)	Surface roughness (nm)
16	5	1.296	—
	10	1.594	—
	20	1.595	—
40	5	0.213	1.57
	10	0.235	0.91
	20	0.299	0.87

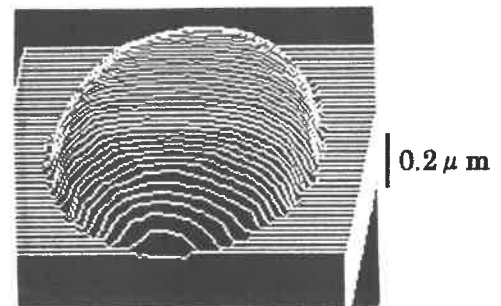


Fig.3 Flatness measurement over the whole wafer surface

4 Machining mechanism of Si wafer polished on BaCO₃ disk

In order to analyze the mechanisms of the mechanochemical polishing of Si wafers by BaCO₃ powder, debris collected after dry polishing experiments was observed and analyzed by SEM, TEM, X-ray and EB diffraction, XPS and AES. Further investigation was carried out by heat treatment experiments of the debris and mixed powders for estimating the reaction process in polishing operation.

4.1 Debris analyses

A 12 mm diameter pellet of 0.4 μ m BaCO₃ powder bonded with 10 wt.% phenol resin was pressed at 100 kPa against a Si wafer rotating at 150 rpm. SEM photographs of the debris collected are seen in Fig.4. The characteristic bar-type debris might be formed as a result of puddling and joining of a lot of debris at the sliding interface. X-ray diffraction analysis of the debris has revealed that almost all debris exists in amorphous state. TEM and EB diffraction observations of the debris support this situation as seen in Fig.5, which further shows that the elements of Ba and Si have been both detected at the same spot by EDAX analysis, indicating the possibility of forming reaction products like barium-silicates.

Figure 6 shows the XPS charts of the debris and the related raw materials. The peak at the binding energy of 103 eV represents oxidation shift of Si 2p, of which position is 99 eV in case of pure Si. From this figure it is clearly recognized that the debris includes barium-silicate, a large amount of silicon oxide and a little unknown stuff at about 96 eV. Another remarkable feature is that no pure Si is remaining in the debris. This means that Si wears as silicon oxide and reaction products with BaCO₃ but never as independent pure Si particles. From further precise analyses of the XPS spectra of the debris, existence of various types of barium-silicate such as Ba₂SiO₄ and BaSi₂O₅ could be supposed.

Figure 7 shows density profiles of the elements of Si, O, C and Ba detected by AES on the polished Si wafers from top-surface to inside. It should be noticed that no barium was detected even on the top-surface of the wafer simply washed by clean water after dry polishing on a BaCO₃ disk and the intensity of oxygen remaining on the top-surface in Fig.7(a) is lower compared with Fig.7(b), the case of a mirror wafer on the market.

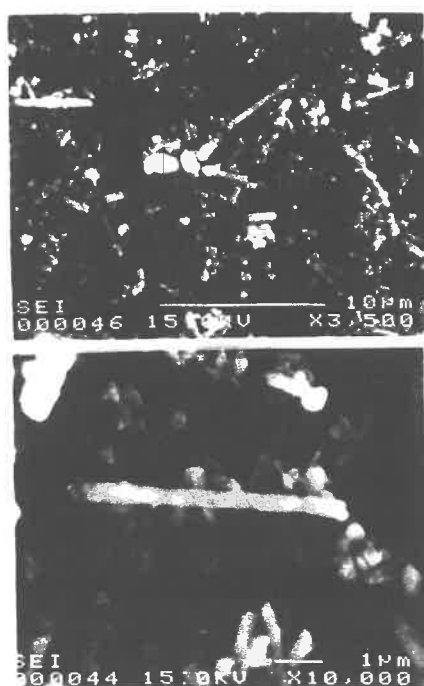


Fig.4 SEM observation of debris

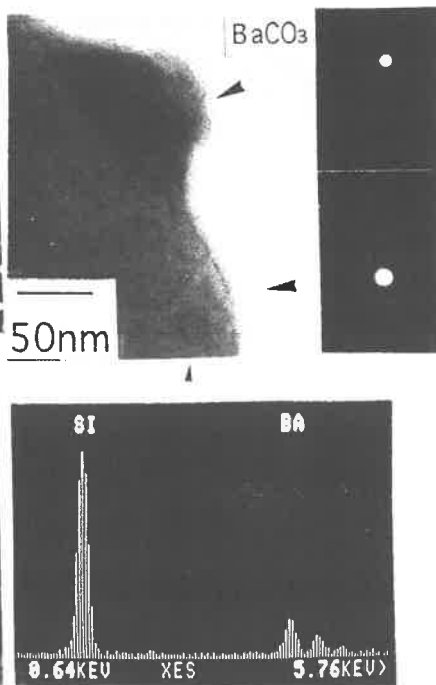


Fig.5 TEM, EB diffraction and EDAX analyses of debris

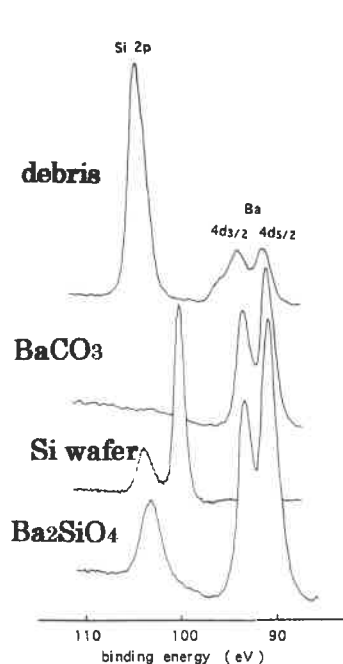


Fig.6 XPS charts of debris and raw materials

4.2 Heat treatments of mixed powder

The debris, which was amorphous in as-received state, was still amorphous by heating at 1000°C. When heated at 1300°C, it changed to form silicon oxide (cristobalite) and barium-silicate (mainly BaSi_2O_5) by crystallization from amorphous phase. This means that both silicon oxide and barium-silicate have already been formed in as-received debris though in amorphous state and this recognition coincides with the XPS analyses of the debris.

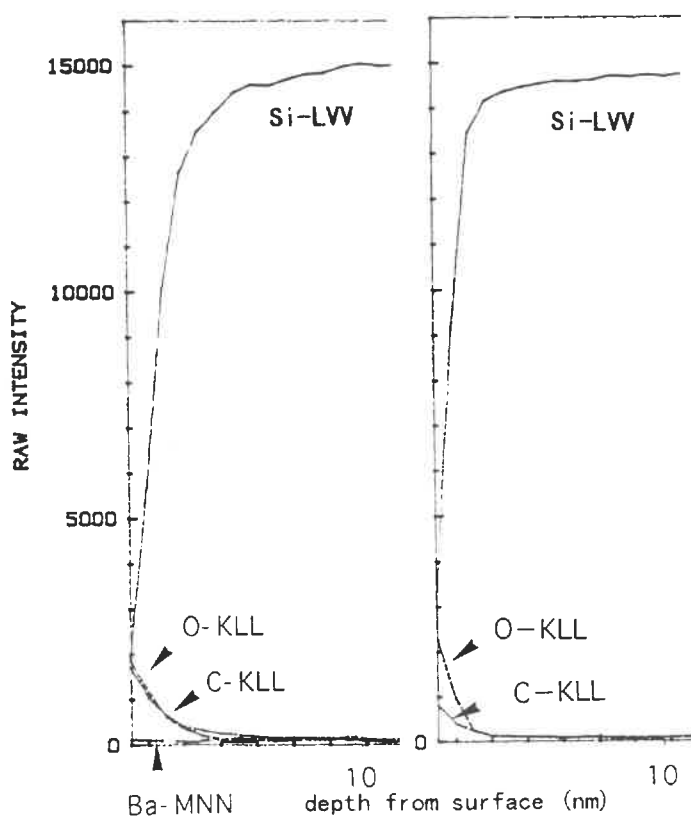
In order to check the possibility of direct reaction of Si itself with BaCO_3 , mixed powder of 95 wt.% Si and 5 wt.% BaCO_3 was heat-treated 2 hours at 1300°C in air. Figure 8 is the X-ray diffraction pattern of the mixed powder compared with that of pure Si powder both after the same heat treatment. It is clearly seen that oxidation of Si powder more proceed in the mixed powder than in pure Si powder. Much higher oxidation of Si in the mixed powder has been observed when the heat treatment was tried in N_2 gas under the same heating condition. These results may prove that BaCO_3 plays the strong role of promoting oxidation on Si surface, which would lead further reaction for forming barium-silicate.

5 Conclusion

- (1) High flatness Si wafers can be obtained by dry polishing on the resin bonded BaCO_3 disk.
- (2) As polishing mechanisms of Si wafers by BaCO_3 powder should be considered promotion of Si oxidation by BaCO_3 powder, subsequent formation of barium-silicate and final dropout of the reaction product in amorphous state.

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(a) BaCO_3 polishing (b) wafer on the market
Fig.7 Depth profile of contamination by AES analyses

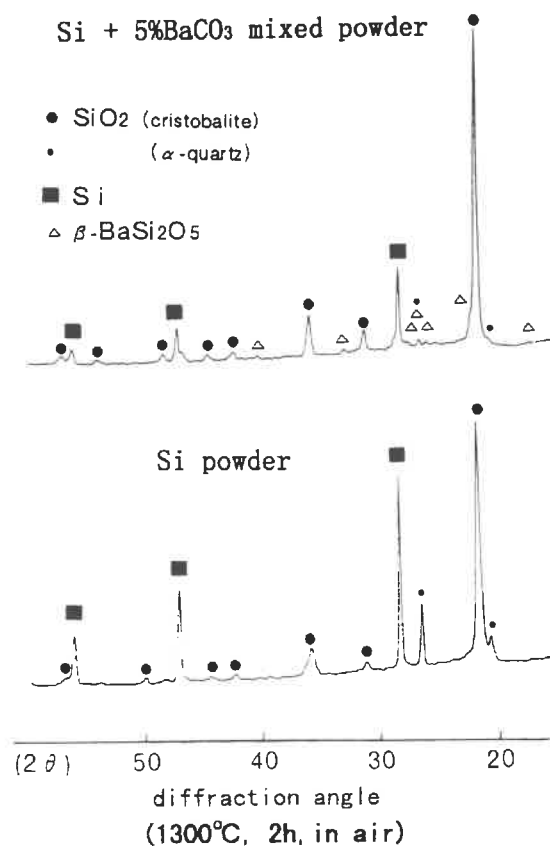


Fig.8 X-ray diffraction patterns of the heat-treated powder

Rapid 3D Microstructure Fabrication by using Excimer Laser Micromachining with Image Processing Technologies

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Abstract

This paper proposes a rapid and precise laser ablation technique for fabricating 3D microstructure with polymer materials by using industrial excimer laser and commercial gray-level image processing technology. The unique character of excimer laser is that the short-duration ultra-violet laser pulses can rapidly breaks the chemical bonds of polymer material within a restricted 'volume'. It means not only the side-wall of the microstructure but also the machining depth can be precisely controlled. In addition, with the advanced image processing technology, a desired microstructure can be conveniently translated to gray-level type image file with a popular format such as .bmp, .gif, .jpg, .dxf, and so on. A NC code generator is proposed to translate the gray-level datum of image to the stage moving path and the laser firing time. The experimental result shows that the proposed technique is a convenient tool for three-dimensional microstructure fabrication.

Key words: Excimer Laser, Micromachining, Gray-level Image Processing.

Introduction

X-ray Lithography technology, which has been demonstrated to be a promising method for the fabrication of 3D microstructure with high aspect ratio. The synchrotron radiation provides the advantages of accuracy potential, high intensity, local divergence, and excellent parallelism, therefore, the extreme precision and high aspect ratio microstructures can be fabricated for special applications in modern technology fields [1]. Although x-ray lithography capabilities are great, few can utilize this process because of lack of access to a synchrotron source and the high fabrication cost. Additionally, the extreme precision of the x-ray lithography process is not required in some applications. Consequently, recently there has been growing interests in low-cost, commonly available processes in which x-ray lithography is replaced by a cheaper alternative technique such as UV photolithography [2], deep reactive ion etching (RIE) [3], or excimer laser micromachining [4].

Excimer laser micromachining is a dry etching process in which deep UV radiation is delivered on the surface of a polymer, and the incident energy is absorbed in a thin layer which is rapidly decomposed, heated and ablated. For the purpose of direct ablative micromachining the most useful wavelengths are 248nm (KrF) and 193nm (ArF). Complex 3D relief may be produced either by scanning a shaped beam over the resist surface, or by projecting a mask to transfer a complex pattern onto the workpiece [4]. Typical characteristics of various deep lithography techniques are showed in Table 1.

Method	Depth (Max.)	Linewidth Control	Cost
Photolithography	$\approx 100 \mu\text{m}$ (Absorption limited)	Poor	Low
Excimer Laser Micromachining	$>200 \mu\text{m}$	Good (μm) at surface	Medium
Photolithography +RIE	$100\text{-}500 \mu\text{m}$ (Etch rate limited)	Good (μm)	Medium
X-ray Lithography	$> 1000 \mu\text{m}$	Excellent (Sub- μm)	Very High

Table 1. Typical characteristics of various deep lithography techniques

We developed a rapid and precise excimer laser dragging machining technology by combining gray-level image process[5]. Through precise controlling fluence of excimer laser, stage velocity and ablation rate of polymer, the gray-level of image will acts as a defined depth. Generally, 256 gray-level is popular and easy translated from any image format. In this paper, we will demonstrate the potential with the concept of combining gray-level as a machining depth to precisely laser machining control.

Experimental Details

a. Machining tools:

Of particular importance is the fact that excimer lasers can remove material by ablation rather than through thermal mechanisms, such as melting, evaporation, or vaporization. This ablative mechanism results in a higher degree of precision than can be achieved with other types of laser processing. Consequently, the microstructure is suitable to be fabricated by excimer laser machining. Excimer laser micromachining were performed by means of a commercial machining

workstation (Exitech series 7000) with beam homogenizer and computer-controlled system for laser firing, beam attenuation, imaged aperture, and mask and workpiece position. Fig. 1 shows the schematic diagram of Excimer laser workstation used in this study. Exposures were conducted by using a Lambda Physik LPX 210 laser operating at 248nm wavelength. The X-Y-Z stage is driven by the DC-servo motors with rotary encoders. X-Y stage resolution and bi-directional repeatability is 1 micron, and maximum speed up to 100mm/sec. Z stage resolution and bi-directional repeatability are 0.1 micron. The 10x projection lens with 0.2 numerical aperture giving fluences up to 10 J cm^{-2} , transfer the mask patterns onto the workpieces. In the excimer laser micromachining system the smallest

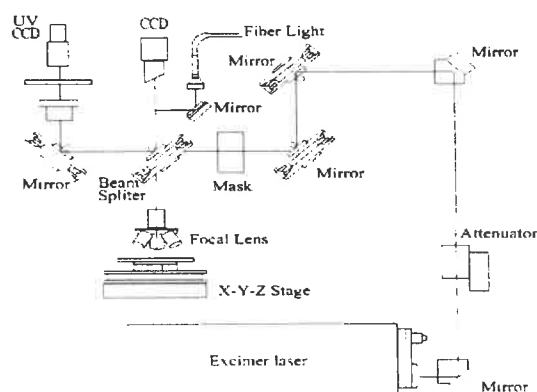


Fig. 1 Schematic of excimer laser micromachining system

lateral resolution is about 1 to 2 μm , and the depth feature is better than 0.1 μm . These system performances will make the quality of the fabricated microstructures steady and reliable.

b. NC Code Generator

NC code generator was developed for translating gray-level datum of image to machining instructions. Two dimensional gray-level image means the gray-level can be treat as the virtual three- dimensional image. The format such as .bmp, .gif, .dxf are standard and popular image format. It is convenient to use, exchange and communicate with advanced image process software. Fig. 2 shows the flow chart of NC code generated from an image file. An image translated by image processing to gray-level image format, then load in NC code generator to generate a machining program with NC code. NC code generator is consisted of excimer laser machining model , optical gain value, aperture size or mask type and ISO (RS274) G&M code.

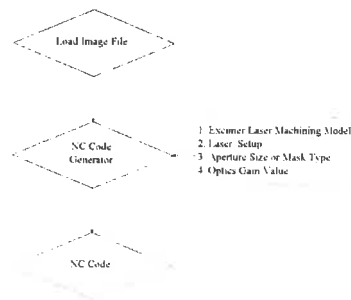


Fig. 2 the flow chart of NC code generator

c. Fabrication Process and result

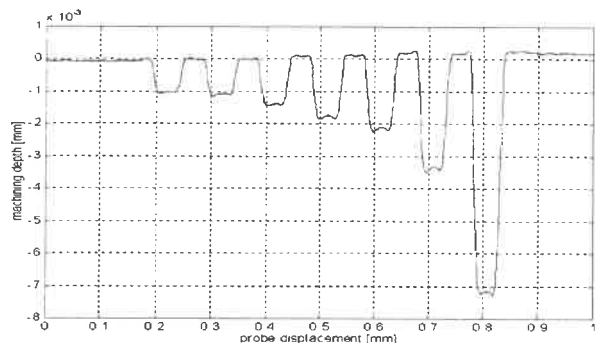


Fig. 3 the cross section measurement result of machining parameter finding

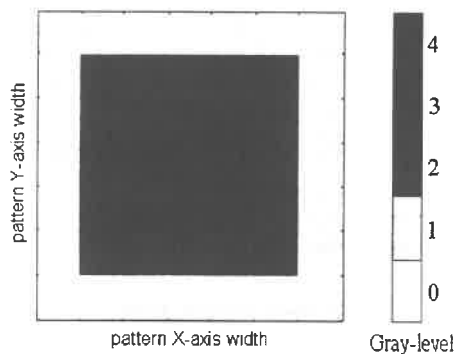


Fig. 4 the sample of 4 level gray-level image

The parameters used in the study of micromachining process are the excimer laser illumination power P , repetition rate H , illumination time within machining period W , attenuator decaying rate U , and magnification of focal lens G . The polymer material used for this machining work is polycarbonate. The parameters in fabrication process are $P=40\text{k Watt}$, $H=50\text{Hz}$, $W=5\text{ms}$, $U=0.3$, $G=10$. A $0.5\text{mm} \times 0.5\text{mm}$ square on the mask is used to be a projection pattern. As shown as fig.3, the sample dragged 2mm with different stage speed 0.125 mm/sec, 0.5 mm/sec, 0.375 mm/sec, 0.85 mm/sec and 0.975 mm/sec. The polymer used in

this process is $150\text{mm} \times 150\text{mm} \times 250 \mu\text{m}$ polycarbonate thick film. In the condition claimed above, the machining depth with fluence on mm square is 16~19 $\mu\text{m/sec}$, and average is 17.5 $\mu\text{m/sec}$. A surface profiler

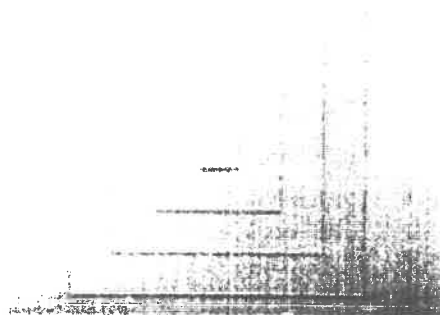


Fig.5 the photography of the machining sample

(Tencor Instrument alpha-step 200) which resolution is 10 nm serves the machining depth measurement.

Fig. 4 is the square sample image with 4 gray-level and 3.5mm width. The darkness of gray-level expresses the machining scan times (0~4 respectively.) responds with defined depth. The moving speed of stage is 0.5mm/sec while laser dragging. The predict depth will be 1.75, 3.5, 5.25, and 7 μ m. The machined result is showed as fig. 5.

For precise fabrication, machining different level

should move the z-axis to keep the projection position. Fig. 6 shows the comparison of result with compensation and without compensation the position of z-axis. The depth of dragging is concerned with laser beam energy, scan velocity of stage, and ablation rate. In same machining condition, it will get same depth. The result shows it corresponds with prediction.

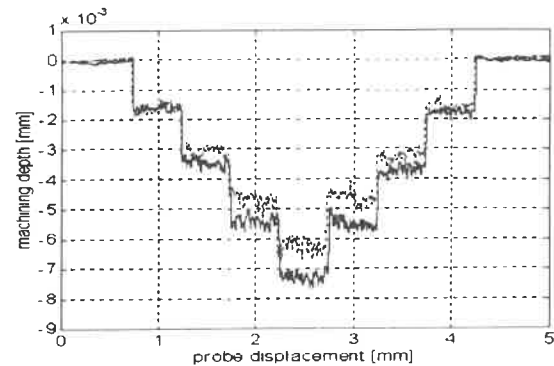


Fig. 6 the comparison of the machining profiles with and without projection plane compensation.

Conclusion

We have investigated the use of polycarbonate thick film to fabricate the three-dimensional microstructures with gray-level image process technology by excimer laser micromachining. The excimer laser micromachining as a complementary tool to other machining processes has been demonstrated to be suitable for fast and precise application. It implies a most possibility by using different pattern should get more complicate and more useful microstructure.

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Evolution of Diffractive Phase Grating Sidewall Slopes

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Abstract

Optical elements known as diffractive phase gratings radiate light in different phases from the surface of the grating. The most critical characteristics of phase gratings are step height, duty cycle, and sidewall slope¹. In theory, vertical sidewalls are ideal for maximizing the energy in the diffractive orders, providing optimal signals from the grating. This paper will discuss the development toward obtaining features in the substrate with nearly vertical sidewall slopes.

Discussion

Pattern transfer is typically done with a resist mask: commonly e-beam, ion beam or UV sensitive resists. This paper will discuss and evaluate pattern transfer exposing features on a substrate using standard semiconductor fabrication processes that utilize a photosensitive film (photoresist), a patterned reticle, and an exposure source (i.e. UV light source). Maintaining a uniform film thickness over the entire substrate yields uniform exposure and optimum pattern transfer from the reticle to the photoresist. While uniform exposure is an important parameter that defines the duty cycle, sidewall slope is of interest in most applications. The primary objective for phase gratings is the ability to achieve nearly vertical sidewall slopes in the final grating material. In the photoresist material, the sidewall slopes can be altered through the use of various processing techniques prior to exposure.

Development work completed at Shipley Company and IBM can be used in an application to alter sidewall slopes for phase gratings [1]. The development work completed at IBM and Shipley was aimed at metal patterning. These groups were interested in developing metal interconnects that were thinner (1 μ m) than the standard 7 μ m line widths. Problems were encountered with typical metal etch techniques, therefore "lift-off" offered an alternative. The lift off process is summarized in the following schematic. Figure 1 depicts the typical steps for photolithography processing. The limitations of this process are mainly the sidewall profiles that result from the exposure and development stages. In Figure 1, these sidewalls are depicted as vertical allowing optimum conditions for lift-off. In reality, the above processing steps would result in sidewall profiles that are not vertical but instead have a positively sloped sidewall.

¹ Sidewall slope is defined to be the angle between the horizontal plane of the substrate and angle measured counterclockwise from that horizontal plane to the feature slope.



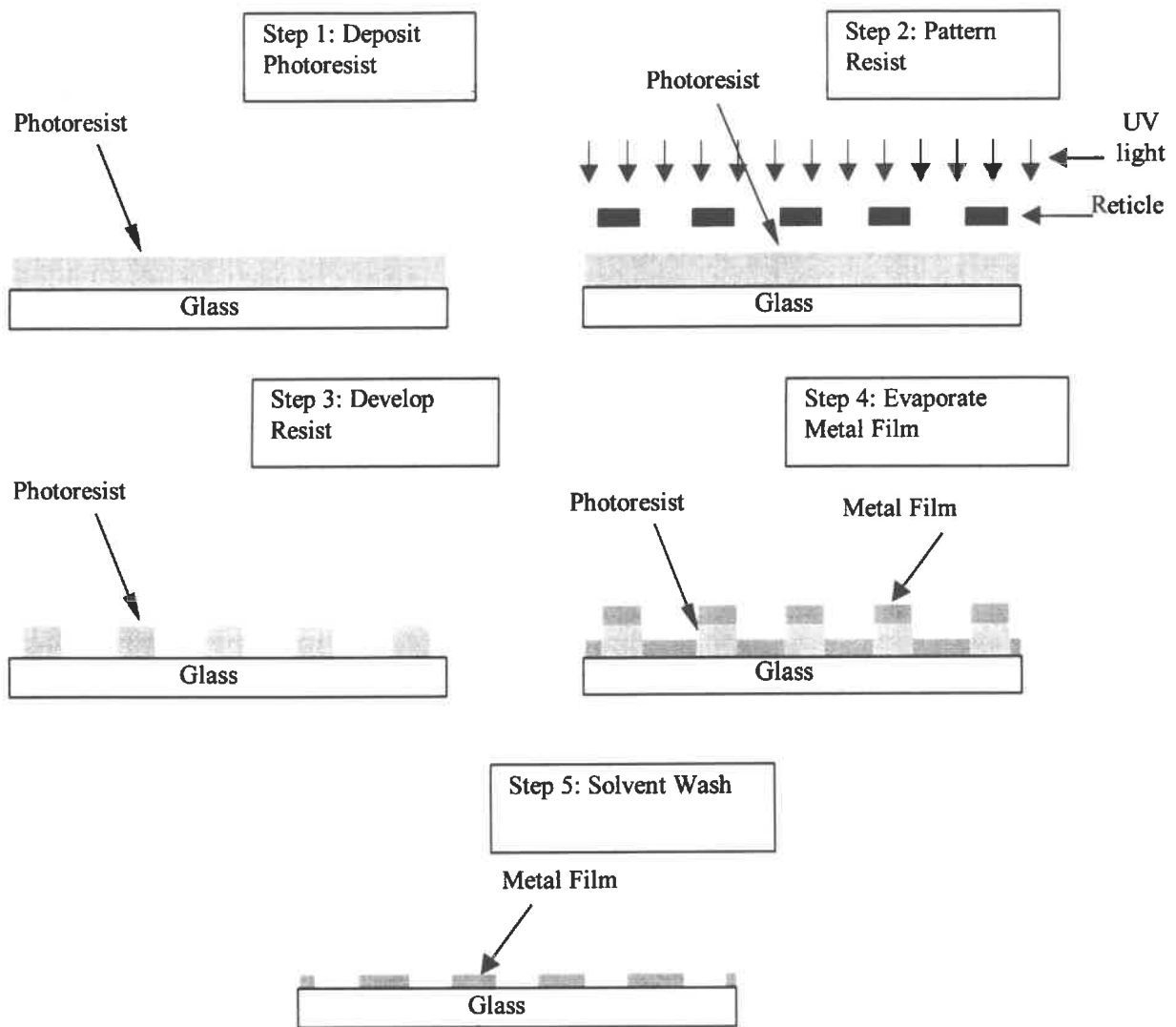
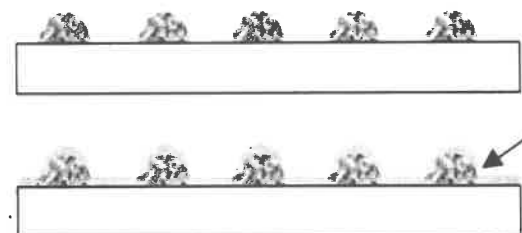
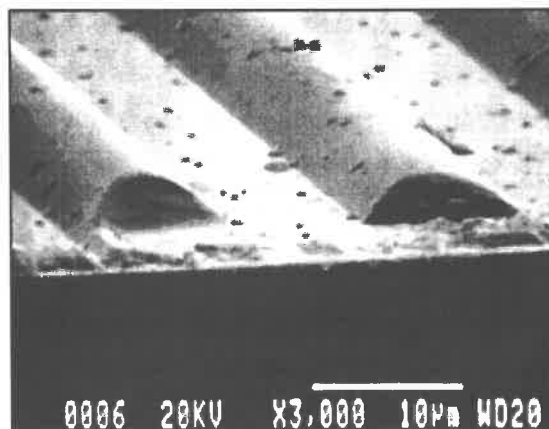


Figure 1. "Lift-off" process steps

Figure 2 shows a SEM micrograph of a typical photoresist profile on a glass substrate. The schematic shows the resulting problem in attempting to "lift-off" of such a sidewall.



Areas where sidewall slope is rounded will not wash away cleanly with solvent wash. Instead, these areas will "tear" and result in a defective part.

Figure 2. Result of a rounded sidewall in photoresist processing.

The limits of patterning were incorporated in the photoresist profiles: the evaporation of metals is a conformal process. The metal film is deposited on top of the photoresist layer, therefore if the sidewall of the resist is positively sloped, the metal film would adhere to the sidewall. During the solvent wash, areas where the profile is positively sloped caused the metal film to adhere to areas that should wash away. The result is tearing the metal mask or the mask not washing away: either result in a defective part.

Work done at Shipley and IBM controlled sidewall slopes by partially developing the photoresist prior to exposure. This process creates a surface layer in the resist that promotes edge definition and assists in controlling the wall slope. The work assessed that low molecular weight components are removed in the “pre-dip”, thus creating a top layer that is less soluble in the final developing stage. The resulting profiles were T-topped, shown schematically in Figure 3.

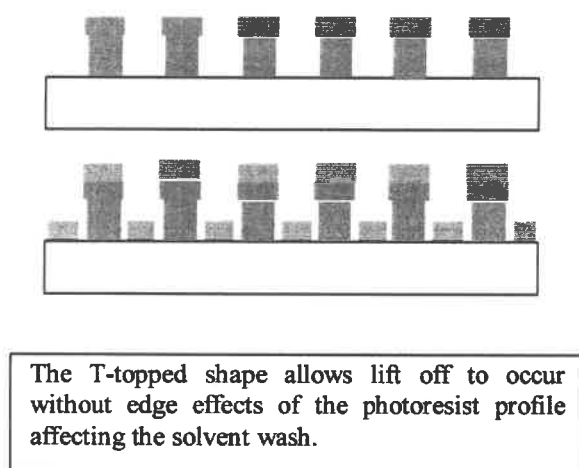


Figure 3. T-topped profile (Shipley Corp)

The T-topped profiles are not the desired shape for phase gratings; however, using the results of this work, an experimental matrix was established to determine the variations in the sidewall slope as a function of the development time prior to exposure.

Experiment

The experimental matrix that was established is shown in the following table.

Table 1. Matrix of Pre-dip Parameters

Sample Number	Pre-dip Solution (% developer in DI water)	Pre-dip Time (seconds)
14	50	120
15	50	90
16	50	60
17	50	30
18	75	30
19	75	60
20	75	90
21	100	30
22	100	60
23	100	90

From Table 1, the pre-dip solution is defined by the concentration of developer in DI water. The results from Shipley used 100% developer solution for pre-dip. The influence of the concentration of developer in DI water was evaluated in an attempt to eliminate the T-topped mask shape. Using the SEM, evaluation of the sidewall slopes was possible, as observed in the following SEM micrographs.

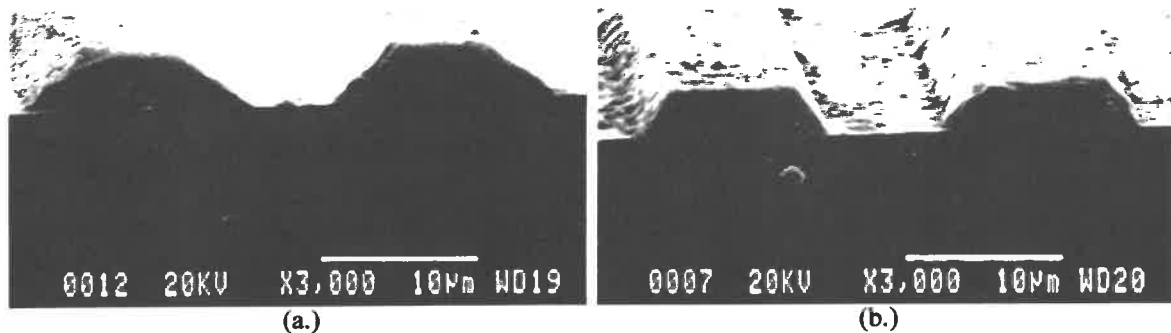


Figure 4. SEM micrographs (magnification 3000 x) (a.) Sample 17 – 30 second pre-dip in 50% developer solution (b.) Sample 14 – 120 second pre-dip in 50% developer solution.

From these micrographs, the diluted developer pre-dip eliminated the T-top shape of the resist. Also, the longer pre-dip produced shallower sloped sidewalls. The influence of hard-baking the pre-dipped features was also investigated and will be presented. The next processing step requires transferring the features directly into the substrate. The main objective is to determine the evolution of the sidewall profile during this processing step. This paper will present the results of research to evaluate the photoresist processing described above and compare the results to other masking techniques using various substrate materials.

Acknowledgements

We would like to thank Anlee Krupp at the Precision Measurement Laboratory at the Boston University Photonics Center for generating the SEM micrographs for MicroE.

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Combined Hardening And Brazing For Dimensionally Stable Assembly

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Keywords: Brazing, heat-treatment, precision assemblies

Technical Areas: Precision fabrication; novel processes

INTRODUCTION

Joining parts permanently in an assembly is an important process in manufacturing. Welding and brazing are important joining processes for assembling parts, as they can provide joints with higher stiffness and lower hysteresis than other assembly methods. Heat treatment is used to provide desirable mechanical properties. Unfortunately, welding and hardening generally distort the assemblies; this requires a final grinding operation to bring the assembly to the desired dimensions. Graph-Air® (AISI A-10) is an air-hardening, graphitic tool steel which is relatively easy to machine in the annealed state. When hardened, it experiences minimal dimensional distortion. This minimizes the time required in post-heat treatment grinding. This makes A-10 a popular material for many precision engineering applications, such as hydraulic servovalves and cold-work dies. Unfortunately, A-10 is difficult to weld.

We have developed a procedure which combines brazing and hardening of A-10; this makes it possible to join A-10 and harden it at the same time. The hardening temperature is high enough to cause the braze material to melt and flow into the joint, while the tempering temperature is below the braze melting point. The combined braze and hardening procedure follows the typical heat treatment cycle: High temperature normalizing, followed by quenching, followed by lower temperature tempering. No additional furnace processing is required. We can therefore design complex assemblies to use oven-braze joints, and combine brazing with hardening. The advantages to this process are the improved stiffness, reduced hysteresis, and shorter assembly cycles, compared to bolted or pinned assemblies, while maintaining the mechanical properties of hardened tool steel. The general process of combined brazing and hardening may also be applied to other materials beside A-10, and depends on selecting appropriate brazing materials compatible with the heat treatment cycle for the selected steel. This paper presents experiments conducted to investigate the joint properties of A-10 using this combined brazing and hardening procedure, including wetting of the base material to the braze material, and the joint strength in shear and in tension.

MATERIALS, METHODS, AND RESULTS

The initial procedure was tested using EZ-Flow 45, a cadmium containing silver braze material. Since cadmium is toxic, we chose to investigate the performance of a joint using the cadmium-silver braze alloy, and compare with a joint using cadmium-free silver braze alloy. The two braze alloys were manufactured by Engelhard: Silvaloy 45 (AWS BAg-1; 45% silver, 15% copper, 16% zinc, 24% cadmium) and Silvaloy A45 (AWS BAg-5; 45% silver, 30% copper, 25% zinc). We used 0.032 inch diameter braze wire. The Timken Company donated Graph-Air™ A-10 tool steel, as annealed bar stock. The flux used was Engelhard Ultra-Flux, a borate-fluoride paste flux.

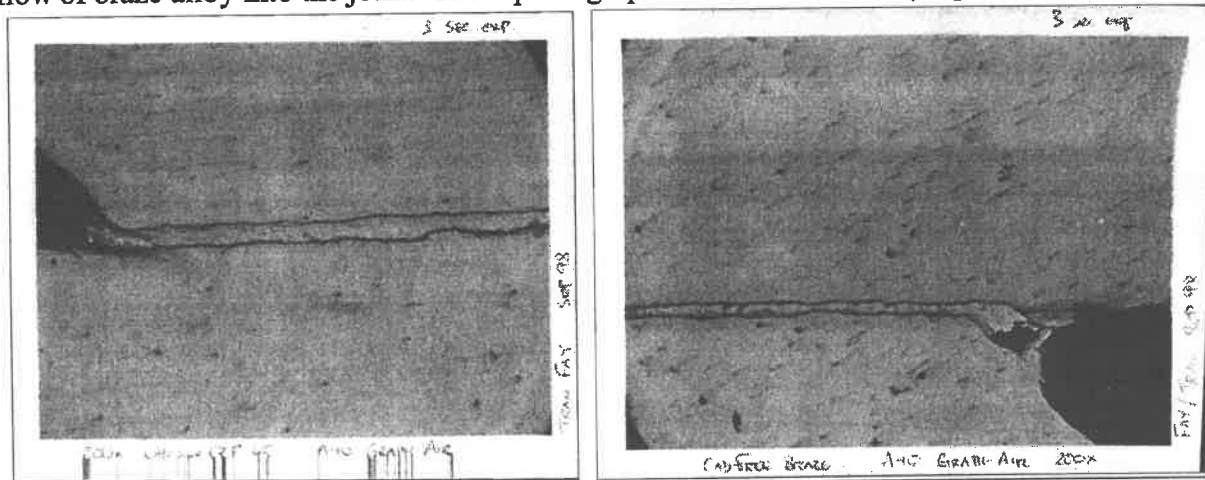
Tensile and shear specimens of A-10 were machined for testing in an MTS tensile test machine. The specimens were designed to test uniaxial tension and uniaxial shear. Flux was brushed onto the contact surface of each specimen. We assembled the specimens, and placed silver braze alloy wire on the edge of the joints; 2 tensile and shear specimens received the Silvaloy 45, and 2 tensile and shear specimens received the Silvaloy-A45. The specimens were then heat treated and normalized to 800°C, using a heating rate of 20°C/min, and soaking at 1 hour at 800°C in an Argon atmosphere. The specimens were removed from the oven and air-quenched. The specimens were then tempered at 225°C for 2 hours.

The specimens were cleaned, then, mounted into a uniaxial Materials Testing System (MTS) tensile tester; the load cell capacity was $\pm 250\text{kN}$. Specimens were pulled at a constant strain rate of 0.2 in/min (0.085 mm/s). Force measurements were recorded at a 10 Hz sampling rate, along with the extension of the specimens. The force to failure was recorded as the peak force reading.

Based on manufacturer literature, we expected both brazing compounds to exhibit tensile strength on the order of 345 MPa (50 kpsi), and using Von Mises criterion, a shear strength of 199 MPa (29 kpsi). We were uncertain if the silver-braze compounds would bond equally well to Graph-Air™ A-10, since brazing was not a recommended joining operation for this material. The tensile and shear test results are shown in the table below:

	Tensile Strength in MPa (kpsi)	Shear Strength in MPa (kpsi)
Cadmium-containing braze	370 ± 3.5 (53.7 ± 0.5)	58.2 ± 27.1 (8.4 ± 3.9)
Cadmium-free braze alloy	410 ± 35 (59.5 ± 5.1)	72.2 ± 1.1 (10.5 ± 0.2)

Photographs of the cross-sections of a brazed joint were also taken, to check the capillary flow of braze alloy into the joint. These photographs are shown below (originally 9cm×12cm).



The left photograph is a 200× photograph of a polished cross-section of the braze joint (cadmium-containing alloy); right photograph is of a joint with cadmium-free braze alloy.

CONCLUSIONS

A combined hardening/brazing operation using Graph-Air A-10 is feasible; with hardening temperature above the liquidus of the braze alloy and tempering temperature below the liquidus. A cadmium-free silver-braze alloy wets the joint as well as the cadmium-bearing alloy. Braze joints should be designed with both shear and tensile bearing features, to take advantage of the superior strength of the tensile joint. A-10 exhibits minimal dimensional distortions after heat-treating, so this combined brazing/hardening procedure can reduce manufacturing steps.

APPLICATION POTENTIAL OF PHOTSENSITIVE GLASSES

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Abstract

Glass becomes an increasingly important material in microsystem technology. Especially glasses, that can be structured by a modified photolithographic process, are significant. Using such a material it is possible to produce any 2 ½ dimensional structure. The photo-structuring process with its steps UV-lithography, thermal treatment and etching supports the production of micro-structured glass components. In order to produce complex objects, e.g. components for microvalves or microactuators, it is necessary to join glass with glass or various other materials. Essential applications of micro-structurable glasses are deformation elements, glass/metal compound coils, fluid components and optical components.

Material and Technological Basics

All used glasses were developed and produced at the Technical University of Ilmenau [1]. Glasses from the $\text{Li}_2\text{O}-\text{Al}_2\text{O}_3-\text{SiO}_2$ system are modified by micro additions, to achieve photosensitivity. Important doping agents are the oxides of Ag, Ce, Sn and Sb. The developed glasses have thermal expansion coefficients of $\alpha_{20-400^\circ\text{C}} = 9.8 - 13.6 \cdot 10^{-6} \text{ K}^{-1}$.

The structuring of photosensitive glass is based on different etching rates of exposed and unexposed glass. The first step is a defined UV-exposure of the glass. A thin chromium layer on vitreous silica is used as a mask, which lays on top of the glass wafer. After the UV-exposure a thermal treatment follows. During the annealing the exposed areas crystallize. The crystals can be etched away with an anisotropic etching process in hydrofluoric acid.

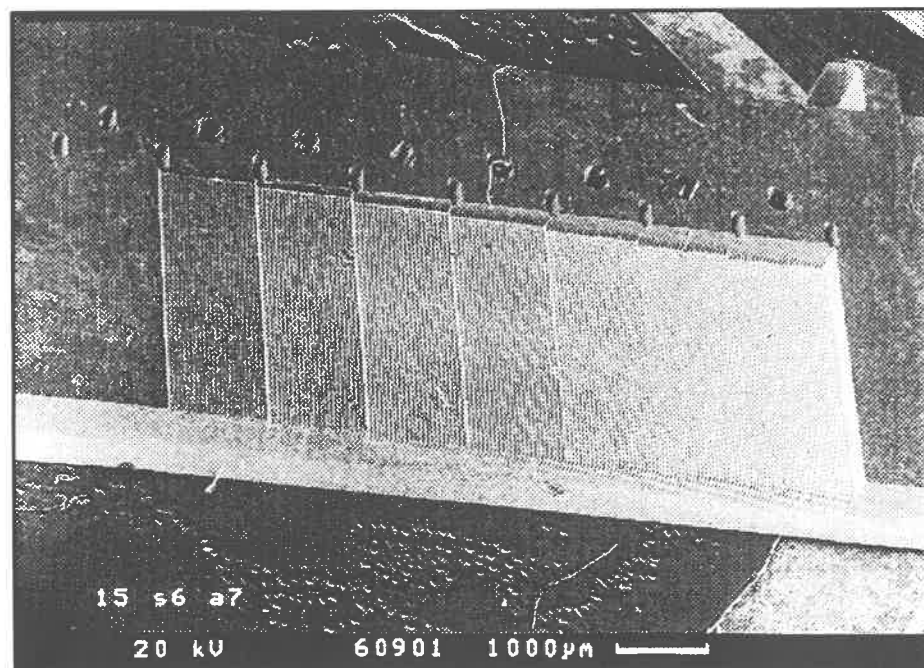


Fig. 1 Depth of etched structures depending on exposure density

Up to now it is possible to etch through the whole wafer or in a defined depth, but not with the same etching process. A new technology [5], using a special technique to realize gray scales-exposures, allows the 3D etching process in glass. A lot of lines with a defined width and pattern are made in a chromium layer on vitreous silica. A pattern of different gray scales can be produced with that technology. The

gray scale mask results in different intensities of UV-light during the exposure. Because of the different light absorption, the exposure of different areas is in various depths. The advantage of this technology is, that structures can be produced not only with a defined depth but also with a defined design. The gray-scale mask technology opens new possibilities of 3D structuring of photosensitive glasses.

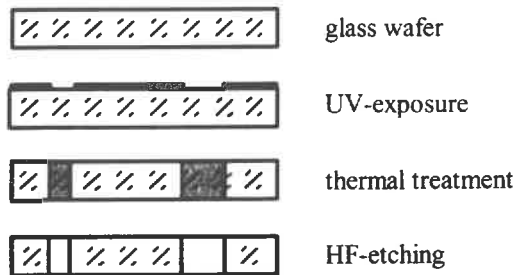


Fig. 2 Standard technology

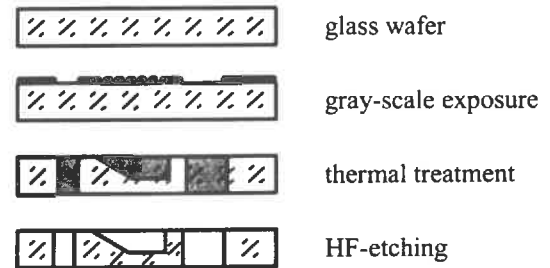


Fig. 3 Gray-scale technology

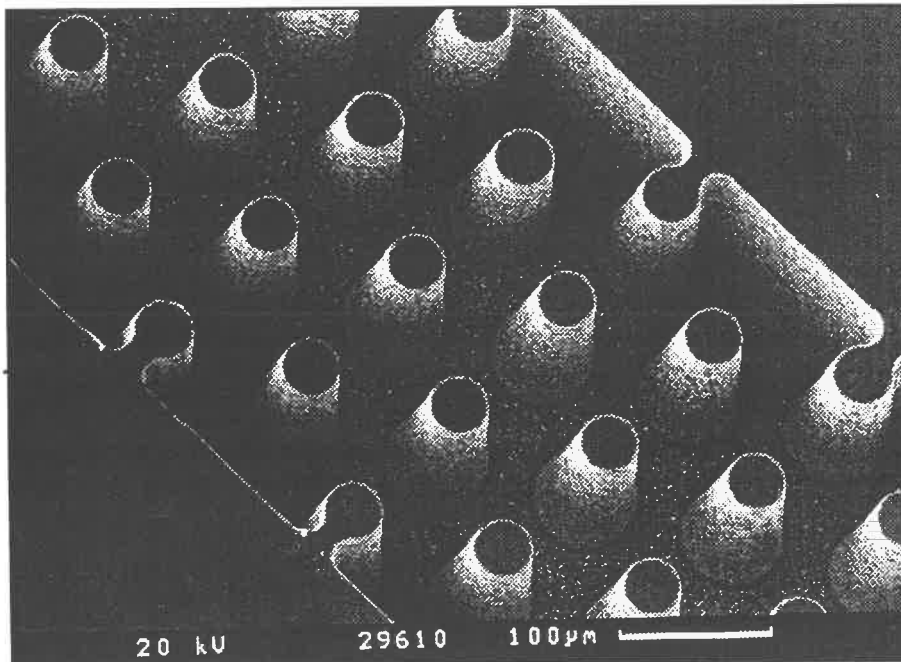


Fig. 4 Channel for a fluid device

Depending on the glass wafer thickness, holes and grooves of 10 µm up to several millimeters width can be fabricated. The slope angle of the walls is approximately 2-6 degrees. Structures with a repeatable precision of 5 µm can be produced. Glass wafers of 0.5 mm up to 2.0 mm thickness can be handled well. At present, the size of the pattern area is limited by the size of the wafer and the maskaligner to 3 inches.

Applications

Essential applications of micro-structurable glasses are deformation elements and elastic glass structures, glass/metal compound coils, fluid components and optical components. Because glass has a good chemical resistance, applications in medicine and chemistry are possible.

Conductors with large cross-section can be produced from etched structures if copper is filled in the glass structures by electroplating [2]. These coils (Fig. 7) can be used e.g. for electrodynamic microdevices [3] [4].

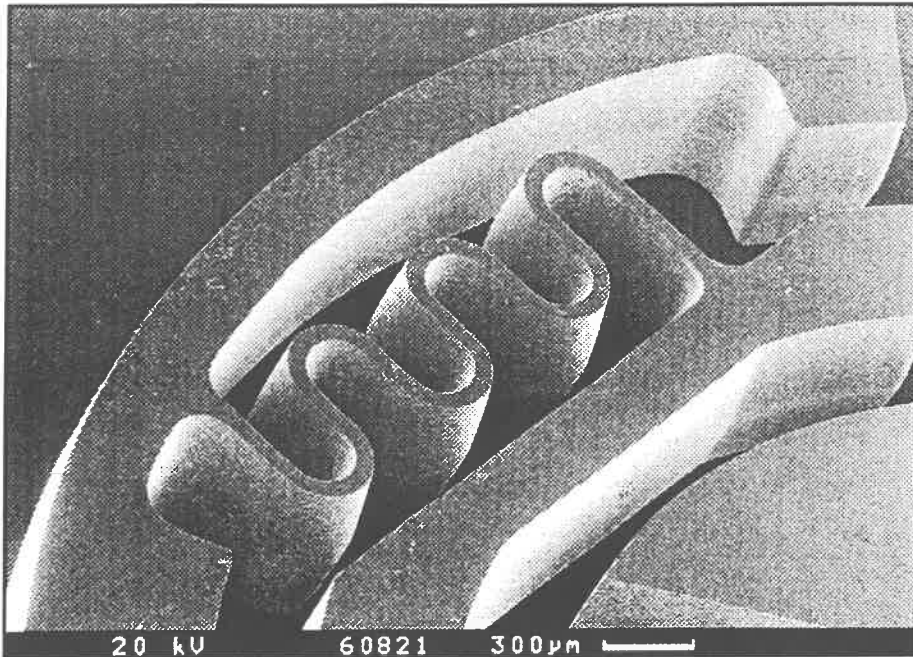


Fig. 5 Elastic glass element (Design: Patent DE 19647579.1)

A high aspect ratio (0.1 mm width to 1 mm depth) and excellent elastic properties (ideal elastic behavior, isotropic material characteristics) make glass an interesting material.

Because circular outlines can be made, springs without peaks of mechanical stresses become possible. An advantage of the etching process is that micro-cracks in the glass are generally stopped by acid polishing.

A micro-force-sensor is a practicable application of photosensitive glass. The force is measured by the displacement of a sensor-spring with linear spring characteristic. In order to sense forces in the field 10-100 μN very sensitive springs with constant material characteristics are required.

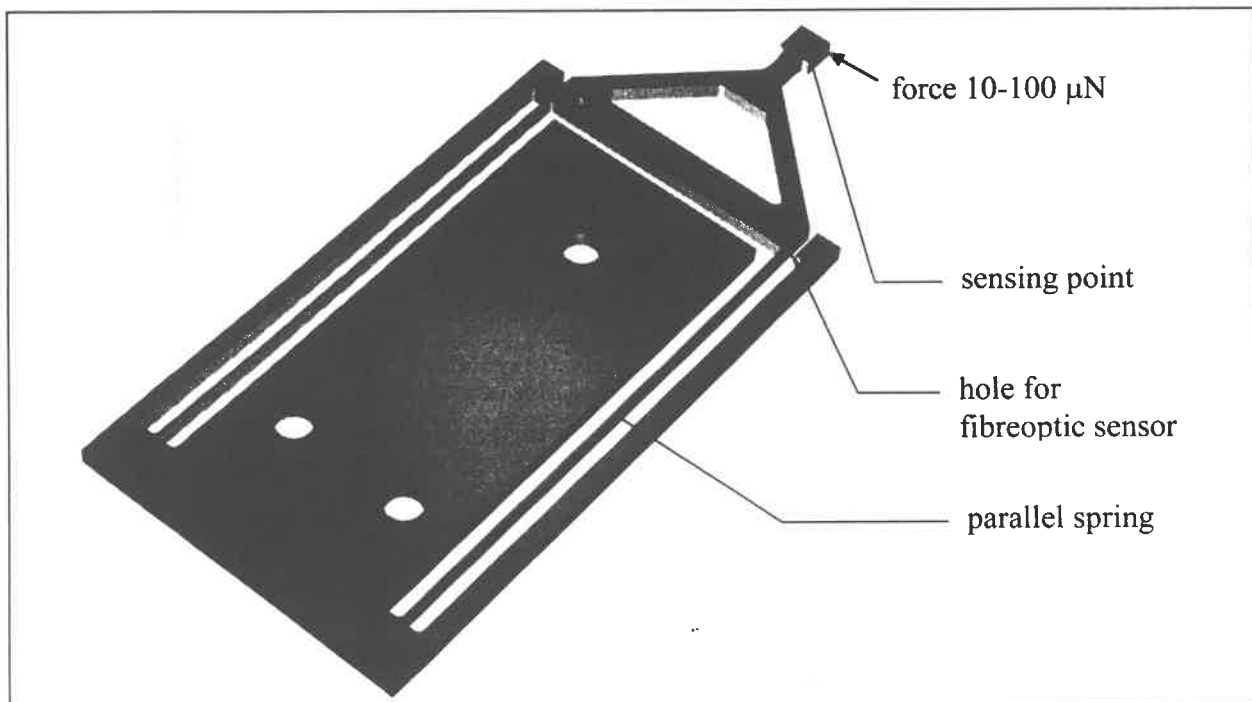


Fig. 6 Glass spring for a micro-force-sensor

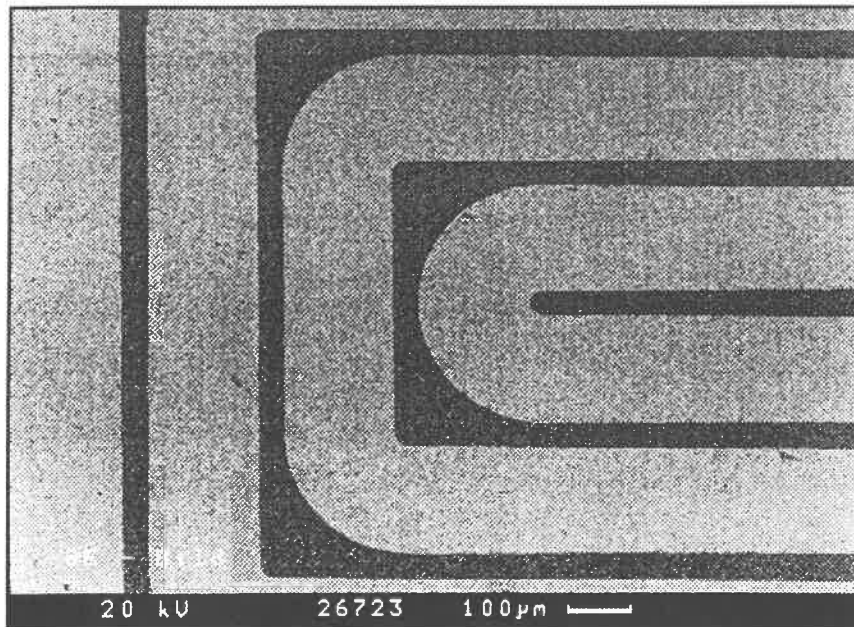
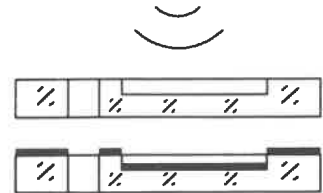


Fig. 7 Electroplated copper coil in glass (glass ← dark)

Microelectroforming in structured glass

Evaporisation



Electroplating of copper



Mechanical treatment



Conclusion

The structuring of glass produces excellent aspect-ratios. Related to other structuring technologies (e.g. LIGA), the glass structuring is inexpensive. All probes were made under normal room conditions. Because of this the glass technology is well suitable for small and medium companies.

Acknowledgments

This work is supported by the German Bundesministerium für Bildung und Forschung (BMBF) under Grant No. 03N1013B0 since March 1996. Special thanks are due to Prof. E. Kallenbach, Prof. D. Hülsenberg, Alf Harnisch, other technical staff of the Technical University of Ilmenau, and O. Mollenhauer from TETRA GmbH.

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ATOMIC SCALE CUTTING PROCESS – 3D MD ANALYSES AND SPM OBSERVATION –

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1. INTRODUCTION

Molecular dynamics (MD) simulation is a useful tool for understanding the atomic scale direct cutting processing. Three-dimensional (3D) MD analyses can reveal the atomic scale surface roughness of workpiece after the cutting process although analyses need much calculation time. The objective of this paper is to clarify the effect of temperature and interatomic force between workpiece and tool on the atomic scale cutting mechanism and surface roughness, by mean of 3D-MD simulation. Assuming that the interatomic potential between the workpiece and tool is derived from the Morse potential function, 3D-MD cutting simulations were carried out with changing temperature and the values of Morse potential parameters r_0 , D and α . The effect of tool materials on the cutting mechanism was also discussed by 3D-MD analyses. The interatomic force between tool and workpiece used here was assumed by the two-body interatomic potential function based on the *ab-initio* molecular orbital calculation. Analytical results were compared with the nano-scale cutting experiments with AFM.

2. ANALYTICAL PROCEDURE

For examining the effect of temperature and interatomic force between workpiece and tool on the atomic scale cutting mechanism, MD simulations were carried out for a nickel single crystal workpiece with a (111) top surface. To reduce the calculation time, the interatomic force of nickel workpiece was assumed to be a pair potential function proposed by Milstein [1]. The Morse potential function as an interatomic force between the workpiece and tool was also assumed since the function has a simple form. MD cutting simulations were carried out by changing the constants of Morse potential parameters r_0 , D and α . In these analyses, values of r_0 , D and α were changed from 1.8 to 2.4Å, 0.01 to 0.15eV and 2.2 to 4.2Å⁻¹, respectively. The temperature of the system was kept constant in the temperature range of 300K-673K by Nose's method [2] to examine the effect of workpiece temperature on the cutting mechanism.

To reveal the effect of tool materials on the cutting mechanism, 3D-MD cutting simulations were also carried out for chromium (Cr) and nickel (Ni) thin film models using diamond and silicon pin tools. Here, the interatomic force between tool and thin film was assumed to be derived from the two-body interatomic potential using parameters based on the *ab-initio* molecular orbital calculation for a (Cr,Ni)-(C,Si)₆H₉ atom cluster. The interatomic force of Cr, Ni thin films and tip tools was assumed to be FS, EAM and Tersoff potential function [3-5], respectively. The machinability of the atomic scale cutting was evaluated with

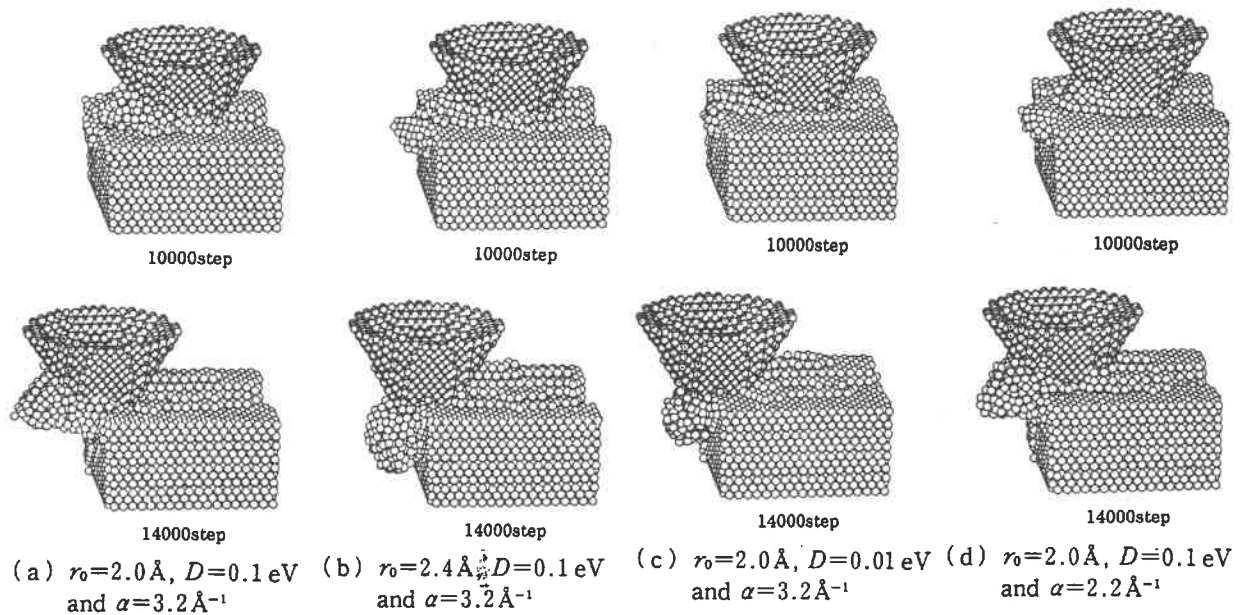


Fig. 1. MD analyses at each interaction between workpiece and pin tool

the ratio of the normal force to the tangential force; F_n/F_t , where the increasing F_n/F_t results in the positive effect of machinability and the decreasing F_n/F_t conversely the negative effect.

3. RESULTS AND DISCUSSION

Figures 1(a)-1(d) show the atomic configuration during the cutting process at 10000 and 14000 steps for (a) $r_0=2.0 \text{ \AA}$, $D=0.1 \text{ eV}$, $\alpha=3.2 \text{ \AA}^{-1}$; (b) $r_0=2.4 \text{ \AA}$, $D=0.1 \text{ eV}$, $\alpha=3.2 \text{ \AA}^{-1}$; (c) $r_0=2.0 \text{ \AA}$, $D=0.01 \text{ eV}$, $\alpha=3.2 \text{ \AA}^{-1}$; and (d) $r_0=2.0 \text{ \AA}$, $D=0.1 \text{ eV}$, $\alpha=2.2 \text{ \AA}^{-1}$. Wedge shaped chips are formed at 10000 steps for all the cases in front of the pin tool. The side flow is also formed at the side of pin tool for every case, which cannot be observed in 2D analysis. Figures 2(a)-2(c) show the atomic configuration at 13000 steps at 373K, 473K and 673K, respectively. Figures 2(d)-2(f) show the atomic configuration toward the moving direction of the pin tool. The chip volume decreases and the side flow increases with increasing temperature (Figs. 2(a)-2(f)). This is caused by the sputtering of the atoms in the chip to the top surface. The sputtered atoms are released from the atomic connection by decreasing the adhesive force between chip atoms due to the kinetic energy increase. Thus, the increase in workpiece temperature has a negative effect on the surface roughness in cutting processing. Figures 3 shows the trajectories of atoms in the 3rd and 6th layers from the top at 10000 and 11000 steps. Workpiece atoms move in the cutting direction at the front of the pin tool. Dislocations are generated at both sides of the pin.

Figures 4(a)-4(d) show the atomic configuration during the cutting process of nickel and chromium thin film using diamond and silicon tip tool. Wedge shaped chips and side flow are formed for the processing of Cr thin films (Figs. 4(a) and 4(c)), whereas slip formations are produced for Ni films (Figs. 4(b) and 4(d)). Wear of the Si tip tool is generated during the processing of the Ni film using the tip and the depth of cut on the Ni film using the Si tip is the shallowest. The variation of F_n/F_t with increasing

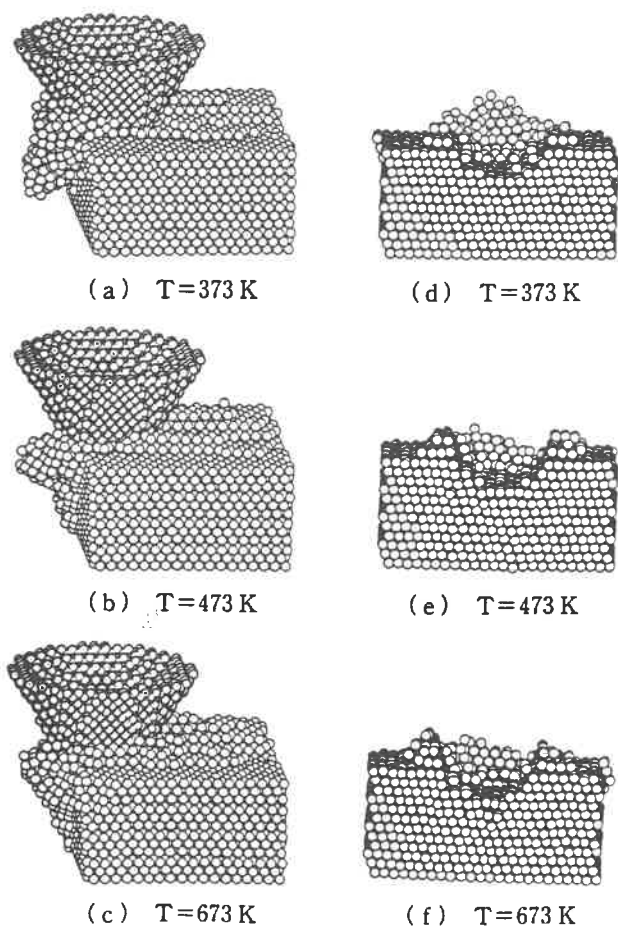
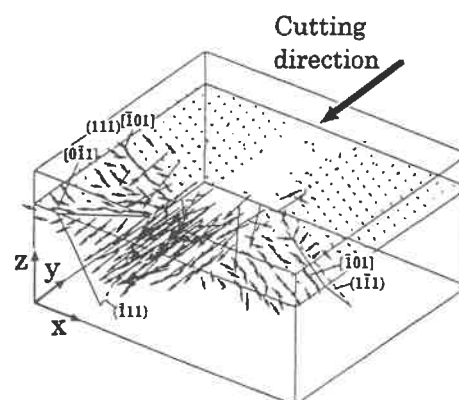
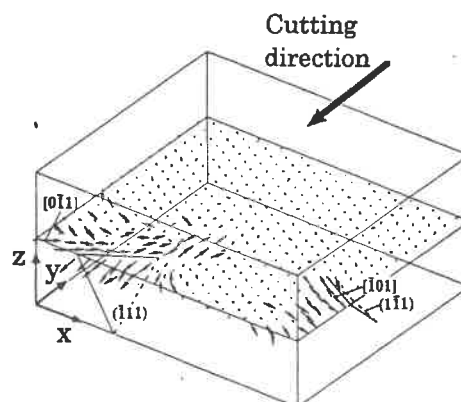


Fig. 2. MD analyses at each temperature



(a) The 3rd layer, 10 000-11 000 steps



(b) The 6th layer, 10 000-11 000 steps

Fig. 3. Trajectory of nickel atoms in 3rd and 6th layer from the top

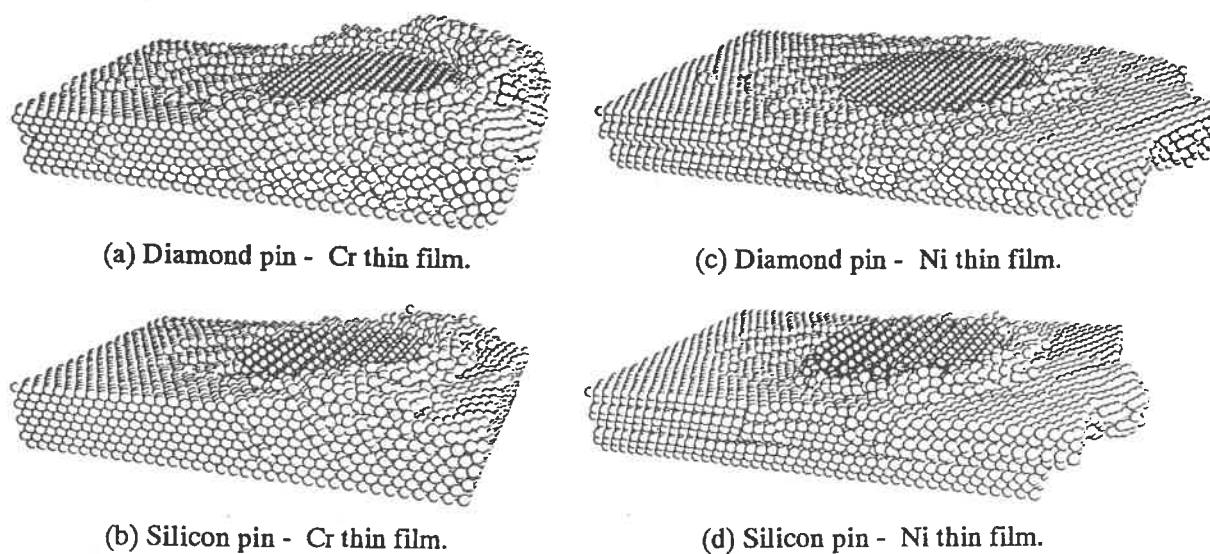


Fig. 4. MD analyses of atomic scale cutting process

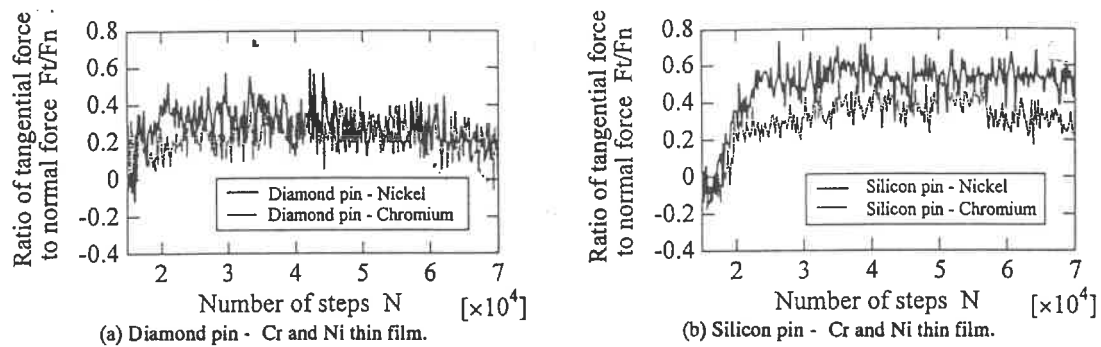


Fig. 5. Variation of F_n/F_t ratio with increasing the number of steps

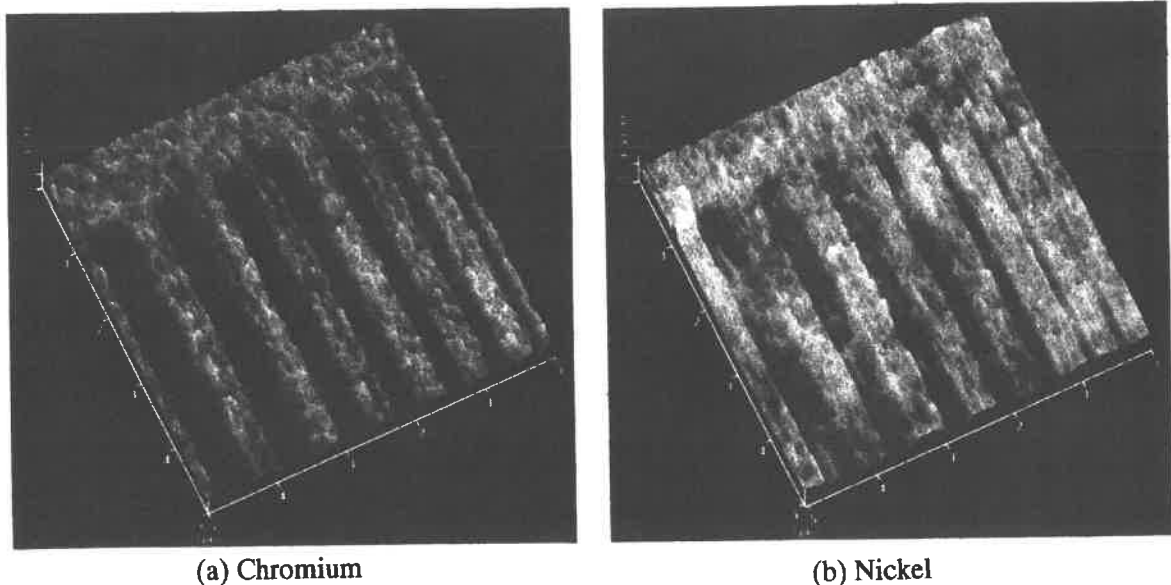


Fig. 6. AFM images of nano scale cutting process

the number of steps is shown in Fig. 5. The ratio F_n/F_t shows 0.2-0.4 during the cutting process of Cr and Ni films using the diamond tip and during that of the Cr film using the Si tip tool, but the F_n/F_t shows 0.4-0.6 in the process of the Ni film using the Si tip. The negative effect of machinability in the process of the Ni film using Si tip was caused by the wear of the tip tool, as shown in Fig. 4(d). Figures 6(a) and 6(b) show the surface of Cr and Ni films scratched by the Si cantilever in AFM. The depth of nano-scale lines formed on the Cr and Ni film surface is about 1.0nm and 0.5nm, respectively. The lines on the Cr film are deeper than those on the Ni film. So, MD analytical results qualitatively agreed with the experimental results.

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Rapid Tooling Using Metal Fiber Reinforced Epoxy

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Abstract

Recently, the application field of rapid tooling technology that manufactures molds using the R/P master model has become wider. Among the rapid tooling, slurry casting generally was performed with alumina powder and phenol. Yet phenol has been replaced by epoxy, because mechanical properties of epoxy are better. Also, all kinds of powder have been used as filler. These powders are expensive and hard to get. While we were looking for the solutions to this problem, it has been found that metal short fiber can be fabricated easily and cheaply, if the self-excited vibration of an elastic tool is used on the lathe. A mold has been created by slurry, mixed with short fiber and epoxy. The case that the powder is only used, was compared with the mechanical characteristics.

In consequence, it has been found that the hardness gets higher, while the shrinkage rate lower, if mixed with short fiber.

Key words: Concurrent Engineering, Rapid Prototyping, Rapid Tooling, Slurry Casting, Al powder/fiber Filled with Resin(AFR)

1. Introduction

Dies and molds need to be produced more cheaply and faster, as the shape of them becomes more complex. To meet the needs, the concept of concurrent engineering has risen and the expectation about the rapid tooling technology has grown as the means of achieving it.⁽¹⁾ One of the rapid tooling methods, the slurry casting mainly used a powder. The powder, however, requires too complex technology to be made and too expensive to use. All of sudden, the thoughts of using chips from a cutting process arose, in order to solve this problem. The chips, however, have too various types to be applied. In the meantime, it has been found that a short fiber can be produced with a large quantity in a while, if the self-excited vibration of an elastic tool is used.

In this study, a short fiber will be applied to the slurry casting method to produce a mold and compared to powder in terms of mechanical characteristics.

2. Production of Short Fiber

An elastic tool, which lacks rigidity as in fig. 1, has been made. Al 7075 was used to make metal fibers. The elastic tool and workpiece are installed on the lathe like in Fig. 2 and it creates self-excited vibration and produces short fibers during a cutting process. Cutting speed has the greatest influence on the diameter of the short fiber, and the length of the short fiber depends on the depth of cut.⁽²⁾ In this experiment, the spindle speed is 1150 rpm and the depth

of cut is 4 mm respectively.

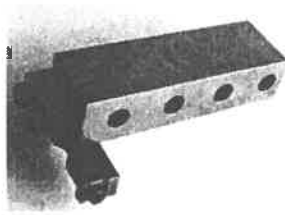


Fig. 1 Elastic tool

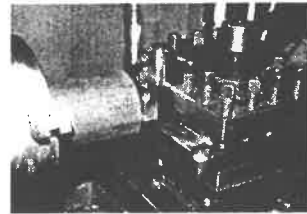
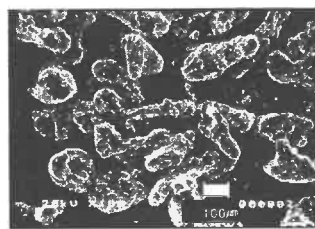


Fig. 2 Operation set up

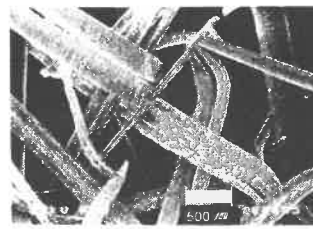
3. Production of Mold

3-1 Materials

The shape of the Al short fiber and that of the powder are shown in Fig. 3. The two materials are blended with an epoxy and turned to a slurry type. Here, it was impossible to produce the slurry, only using the short fiber, since the volume of the short fiber is big, compared to its weight. Fig. 4 represents mixture ratios of slurries.



(a) Al Powder($\times 100$)



(b) Al Fiber($\times 35$)

Fig. 3 SEM photograph of used materials

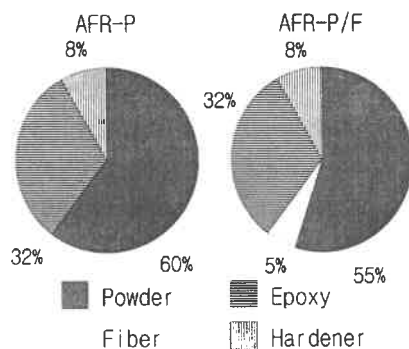


Fig. 4 Mixed ratios of AFRs

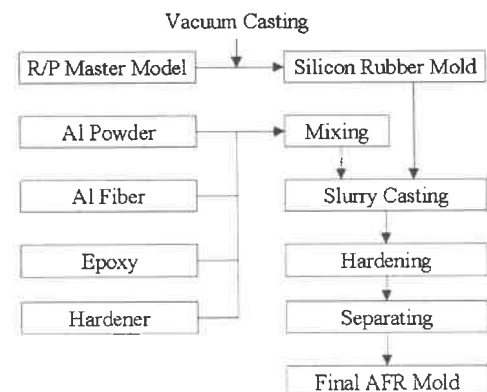


Fig. 5 Manufacturing process

3-2 Process

Fig. 5 shows the whole schematic process.³

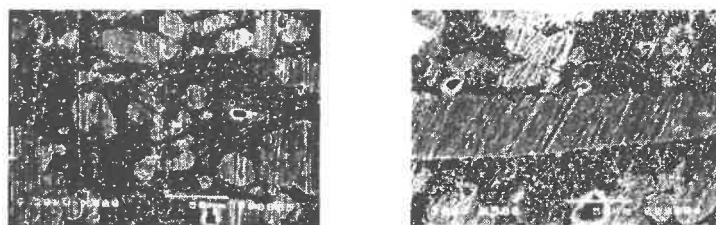
Fig. 6 shows external shapes of the final molds and according to these photographs both of them are perfectly molded. Fig. 7 shows the cross-section of the final molds. The powder is round and white, while the short fiber is long and white. The other black parts are the epoxy.



(a) AFR-P

(b) AFR-P/F

Fig. 6 External shape of final molds



(a) AFR-P

(b) AFR-P/F

Fig. 7 SEM photographs ($\times 500$)

4. Comparison of Mechanical Characteristics

4-1 Shrinkage rates

Table 1 represents the measured shrinkage rates of the final molds. a and b are the width and breadth of the mold. The values of the master model are exact ones, while measured values are average values of the 5 spots. In the Table 1, it is noticed that the shrinkage rate gets smaller if short fiber is compounded with powder. It is due to that the volume of short fiber is bigger than that of powder.

Table 1 Shrinkage rates of final molds

Materials		Exact value	Measured value	Shrinkage rate
a	AFR-P	63.75mm	63.68mm	-0.118 %
	AFR-P/F		63.74mm	-0.016 %
b	AFR-P	120.00mm	119.85mm	-0.125 %
	AFR-P/F		119.88mm	-0.104 %

4-2 Tensile strengths and Elongations

Both the tensile strengths and elongations get smaller when they are compounded with a short fiber, as in Table 2, because the fracture occurs at the point where a short fiber is contained in a transverse way. A research on how to arrange a short fiber vertically in order is required.

Table 2 Tensile strengths and elongations

Materials	Tensile strength	Elongation
AFR-P	43.0MPa	0.723mm
AFR-P/F	31.0MPa	0.463mm

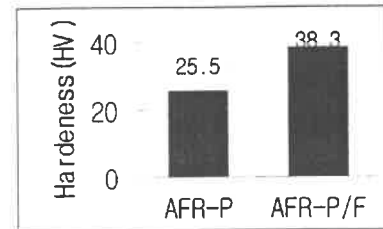


Fig. 8 Vickers hardness of final molds

4-3 Hardness

Fig. 8 represents the hardness value of the final molds. The mean hardness value increases when blended with a short fiber, since the value is high at the point where the short fiber is placed.

4-4 Roughness

Table 3 shows us the roughness of the final molds. The roughness increases when compounded with a short fiber, the short fiber is too wide and long to be blended with powder evenly.

Table 3 Surface roughness of the final mold

(Unit: μm)	R_a	R_q	R_{max}
Master mold	0.040	0.051	0.20
AFR-P	0.119	0.154	0.94
AFR-P/F	0.181	0.224	1.06

5. Conclusions

A short fiber was applied to a slurry casting in this study. It is a good solution to the contemporary problem in Korea. For short fiber can produce a variety of material cheaply only if it can install on the lathe. Also, short fiber will be excellent through mixing process in terms of shrinkage rates and hardness. As for tensile stress, it was expected to be better, compounded with short fiber. The wanted tensile stress doesn't come out because the arrangement of the short fiber is not uniform. The technology should be developed that can arrange it in the way we expect. Also, the technology should be developed that can make rapid tooling possible, only using short fiber, in order to reduce the mold cost.

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Laser drawing system using focused and unfocused beams for manufacturing large diameter zone-plates

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Introduction

The SUBARU telescope, which uses a main mirror with a diameter of 8.2 m, is being built in Hawaii [1]. The mirror has an aspherical concave shape. When general interferometers are used, the following technical difficulties arise in making measurements on the mirror: The influence of the air turbulence and vibrations cannot be disregarded, because the mirror has a large diameter and the optical path length is long. A null lens is necessary when a Fizeau interferometer [2] is used to measure an aspherical mirror. Manufacturing the null lens is difficult. In this case common-path interferometers are useful for the measurement of the surface, and a zone-plate interferometer [3] is one kind of common-path interferometer. A zone-plate interferometer is hardly influenced by air turbulence and vibrations. It is standard practice to use a diffraction wavefront in a zone-plate interferometer. Manufacturing a null lens is more difficult than manufacturing diffraction gratings to generate a reference wavefront. However, manufacturing the zone plate is also not easy. The diameter of the zone plate needed to measure the SUBARU mirror is 100 mm or more. In this case the zone plate is too large to be generated by using the laser pattern drawing machine which was developed by our laboratory [4] and similar machines developed by other organizations [5].

In this paper, an improved laser beam drawing machine for manufacturing large diameter zone-plates is presented. A spherical mirror having a diameter 240 mm is measured by zone-plate interferometer in which a zone plate with diameter of 100 mm is manufactured with the improved drawing machine.

Instrument for manufacturing zone plates

Figures 1 and 2 are a schematic diagram and a photograph of the instrument for manufacturing zone plates. The drawing machine developed by us was improved to make it suitable for generating large diameter zone plates. The dimensions of the instrument are 450*350*200 mm. Precise positioning accuracy is necessary for the writing zone-plate gratings. The laser beam writer is mounted on a sliding table of hydrostatic type with high-precision positioning accuracy. The sliding table travels radially with to the spindle by 0.1 μm according to computer instructions. The moving range of the sliding table is 140 mm.

An Ar laser operating at 458 nm is used as the writing source. The maximum power of the laser is 15 mW. The intensity is modulated by an acoustic-optic light modulator (AOM). The AOM is controlled by a computer model PC-9801DA manufactured by NEC Co., Ltd.

Grating lines are written on a substrate by combination of focused and unfocused laser beams as following process: The first, all outline of the zone is written rapidly by focused beams. And then unfocused beams are used to fill the zone. The time required for generating a zone plate is long if only

a focused beam is used. Furthermore, the profile of the patterns becomes less sharp when an unfocused beam is used alone. Both beams, therefore, are used to improve the accuracy of the zone plates and to decrease the manufacturing time.

Focal position of the beam is determined by measuring the intensity detected by the photo transistor with a pinhole. Unfocusing is performed by a piezoelectric transducer (PZT) manufactured by Polytec PI Inc. controlled by feedback control.

Profile of drawing pattern

To make zone plates with uniform duty ratio, width of drawing pattern is controlled continuously by using unfocusing system. As unfocusing is performed, distribution of laser power has gaussian pattern as shown in Fig. 3. The drawing pattern has a smooth edge. The width of pattern on the resist surface is larger than the width of one at the boundary between glass and resist. The width of pattern drawn by improved machine is measured and is indicated in a graph as shown in Fig. 4. The graph shows that difference between the maximum width and the minimum width increases in proportion to the ideal width of drawing pattern. As the region of slope increases, postprocess for phase control is failed in the manufacturing of zone plates. To be sharpen the edge, a focused beam is drawn at the slope area. The results is shown in Fig. 5. Figure 5(a) shows the pattern drawn by the unfocused beam only. Figure 5(b) shows the pattern drawn by combination of the unfocused and the focused beam. The zone plate is made by this method. The results is shown in Fig. 6.

The zone plate with large diameter was made by using the improved machine. The zone plate was used to measure the spherical mirror with a diameter of 240 mm and a radius of curvature of 170 mm by using the zone plate interferometer. The diameter of the zone plate was 100 mm. Figures 7 and 8 are a schematic diagram and a photograph of the zone plate interferometer. The zone plate put on midpoint between the mirror and its center of curvature. The zone plate is illuminated by plane wavefront. A He-Ne laser operating at 633 nm was used as the light source. Figure 9 shows a fringe pattern obtained by the zone plate interferometer with the zone plate manufactured by the improved machine. A fine interferogram is obtained. The cause for the noise in the fringe is defects of mirror and resist.

Conclusions

The drawing machine developed by us was improved to make it suitable for generating large diameter zone plates. A zone-plate with the large diameter of 100 mm was manufactured by the improved drawing machine using focused and unfocused beams. A spherical mirror with a diameter of 240 mm was measured by the zone plate. A fine interferogram was obtained with the zone plate.

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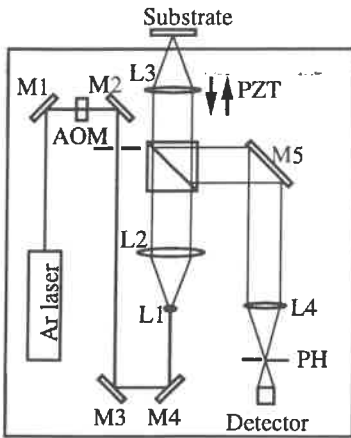


Fig. 1 Scheme of an instrument for manufacturing zone plates.

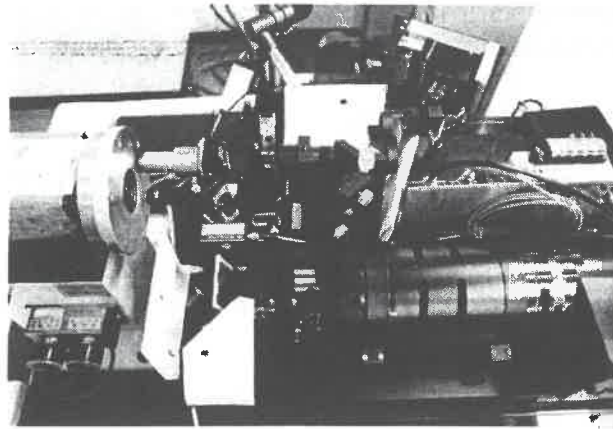


Fig. 2 Instrument to generate zone plates.

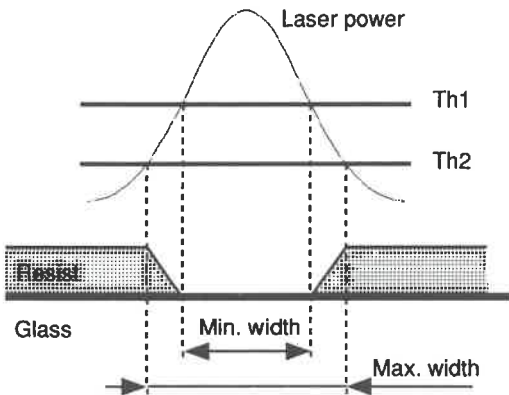


Fig. 3 Profile of the drawing pattern manufactured by an unfocusing system.

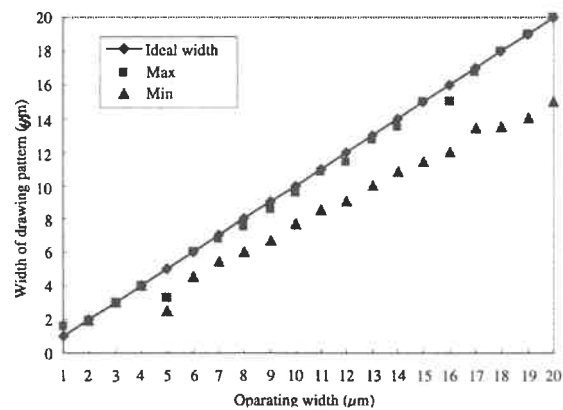
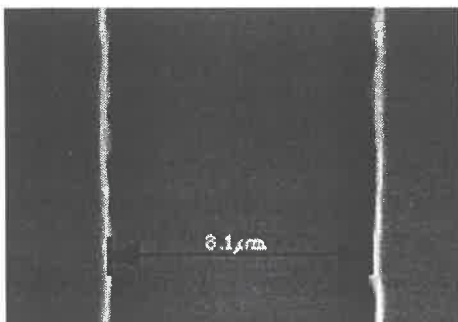
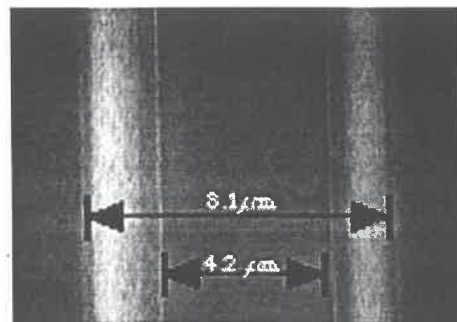


Fig. 4 The relation between the maximum width of drawing pattern and the minimum one.



(a)



(b)

Fig. 5 Experimental results of drawing patterns by using focused and unfocused beams. (a) unfocused beams only, (b) unfocused beams and focused beams.

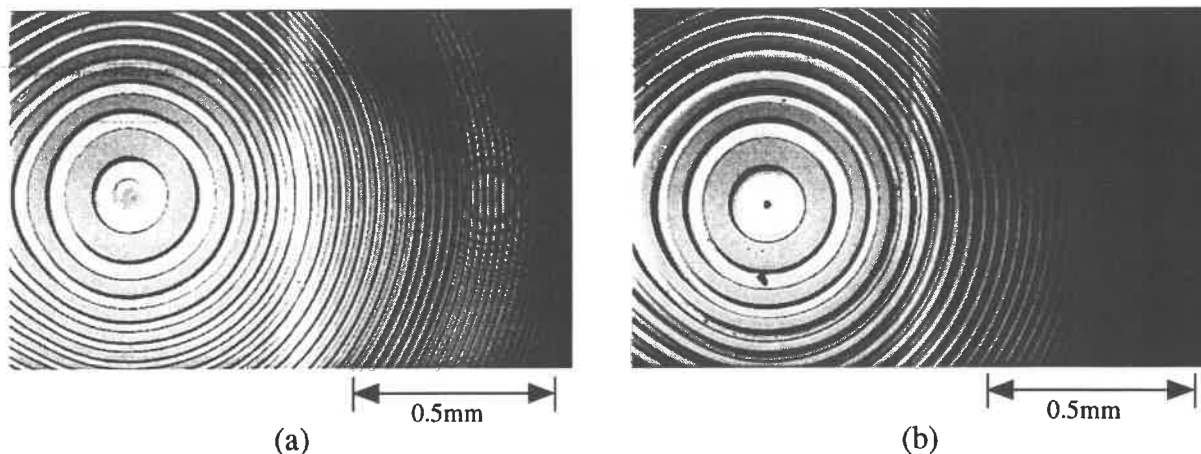


Fig. 6 Comparison of the general method with improved method. (a) general method, (b) improved method.

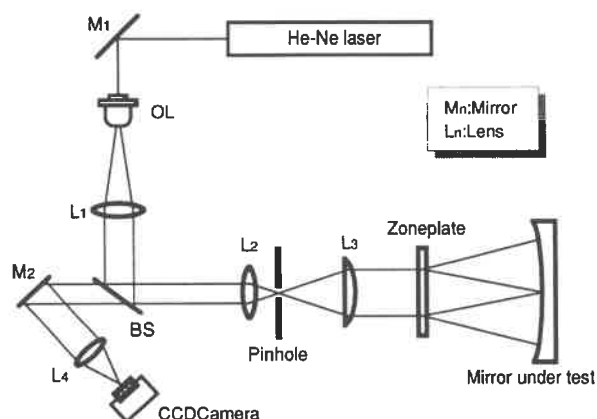


Fig. 7 Scheme of the zone plate interferometer.

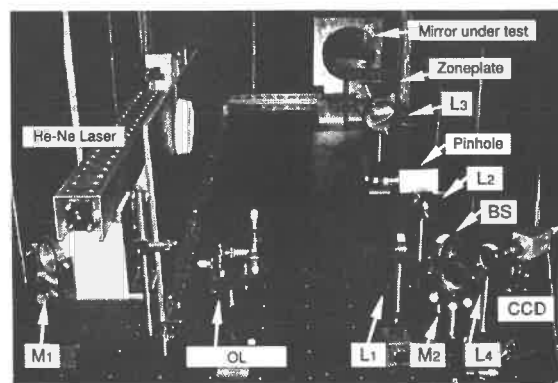


Fig. 8 Photograph of the zone plate interferometer.



Fig. 9 Interferogram obtained by the zon plate interferometer with the zone plate manufactured by the improved drawing machine.

MICRO PARTS AND STRUCTURES FABRICATED BY LASER SINTERING USING POWDER TAPES

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INTRODUCTION

Selective Laser Sintering or layer manufacturing of three-dimensional objects by means of the laser liquid phase sintering of metal has been developed as an aspect of rapid prototyping [1]. Producing parts layer by layer allows a direct coupling with the CAD model of the product in which successive cross sections are calculated. This material processing technology also allows the production of complex parts with complete freedom as to shape and without the need for dedicated tools such as moulds or dies. There can be no doubt that novel rapid prototyping and new fabrication technologies, as well as the rapid prototyping of mechanical components, will hold the key to manufacturing in the 21st century [2].

The present paper describes some further developments of laser micro fabrication using powder tapes [3], the main aims of which are the hybridization of sintered parts and the optimization of sintering conditions. Examples of hybridization and three-dimensional complex objects are shown.

LASER SINTERING USING POWDER TAPE

The authors have developed a new type of layer manufacturing using laser sintering to be applied to the fabrication of micro functional/functional parts [3], as shown in Fig. 1. The novelty lies in the use of a thin tape consisting of micro powders (we call this "powder tape"). A tape (about 200 μm in thickness) made of metal micro powders (about 40 μm in diameter) is prepared beforehand, then placed on a substrate or an existing sintered pattern, and irradiated by an Nd:YAG laser (wavelength 1.06 μm , 0.1-0.2 mm spot size) to form a selectively sintered part. The process is semi-automated and numerically controlled. Powder tapes are easy to handle compared with the loose powder used in the conventional SLS system. Dimensional accuracy is kept in the order of the tape thickness.

Some three-dimensional layered objects have been constructed from powder tapes of copper, nickel and steel, such as tensile test specimens ($4 \times 2.5 \times 20 \text{ mm}^3$) and three-dimensional patterns

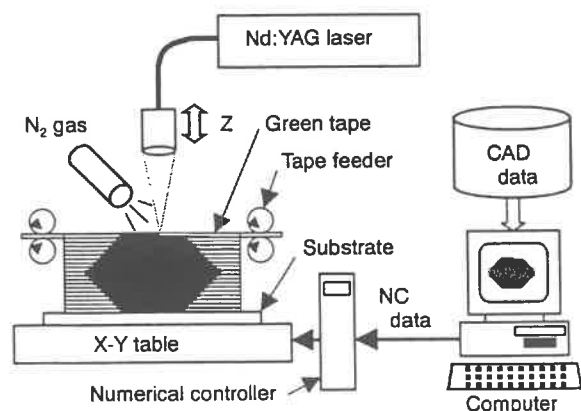


Fig. 1 Laser sintering apparatus using powder tape.

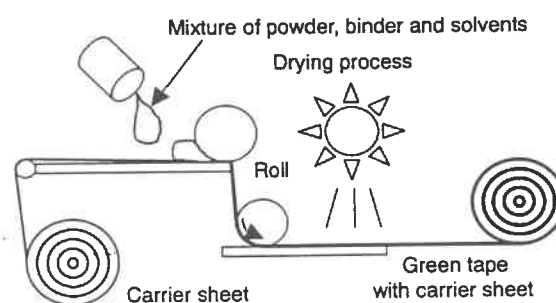


Fig. 2 Schematic of powder tape preparation.

Table 1 Powder tapes and sintering conditions used in the experiment

Tape	Copper	Nickel	Steel
Composition of powder metal	OFHC 99.99%Cu	7%Cr, 3%B, 5%S, 3%Fe, Bal:Ni	17%Cr, 13%Ni, 2.5%Mo, Bal:Fe
Powder size, μm	40	40	40, 100, 250
Powder shape	Spherical	Spherical	Flat
Tape density, g/cm^3	5.3	3.5	4.1
Tape thickness, μm	240	230	210
Beam power P, W	53	45	45
Scan speed U, mm/min	200	150	150
Scan spacing s, mm	0.15	0.15	0.15
Beam diameter d, mm	0.1	0.1	0.1

adhering to a stainless steel plate [3]. The copper object has a density of 4.5 g/cm^3 and a maximum tensile strength of 40 MPa. The feasibility of the proposed method has thus been fully demonstrated. The next stage of the research is to refine the sintering process.

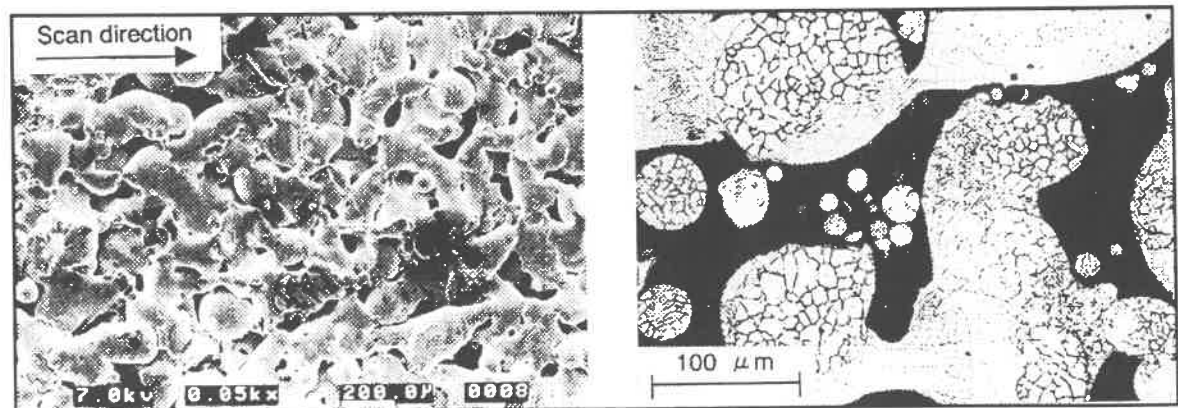
Major improvements planned in the present study are twofold: reinforcement of the sintered part by hybridization, and the optimization of sintering conditions, which enables us to make complicated objects peculiar to the proposed method. A hybrid structure can be made by layering more than one green tape. Combinations of copper, nickel and stainless tapes are tested for formability, mechanical strength and microscopic structures.

Table 1 shows the powder tapes and sintering conditions used in the experiment. The doctor-blade method [4] was employed to cast the green tape. Figure 2 schematically shows the green tape casting process. A slurry with a mixture of metal powder, binder (polyvinyl butyral), plasticizer (dioctyl phthalate) and solvents (toluene, methylethylketone) is spread over a plastic carrier sheet through stainless rollers. A drying process is necessary to volatilize the solvents. The binder contained in the tape should be as little as is necessary to form a powder tape.

THE MICRO PARTS AND STRUCTURES FABRICATED

Hybridization

Figure 3 shows two micrographs of a copper object: (a) top view and (b) section view. Thirteen layers are sintered under the conditions listed in Table 1. Since the sintered part shrinks a little at each layering, more powder is introduced to fill the gap and is pressed on to the green tape. Dimensional accuracy is within the thickness of the tape, i.e. $200 \mu\text{m}$, and no marked warp can be



(a) Top view

(b) Section view

Fig. 3 Copper structures fabricated by the proposed method.

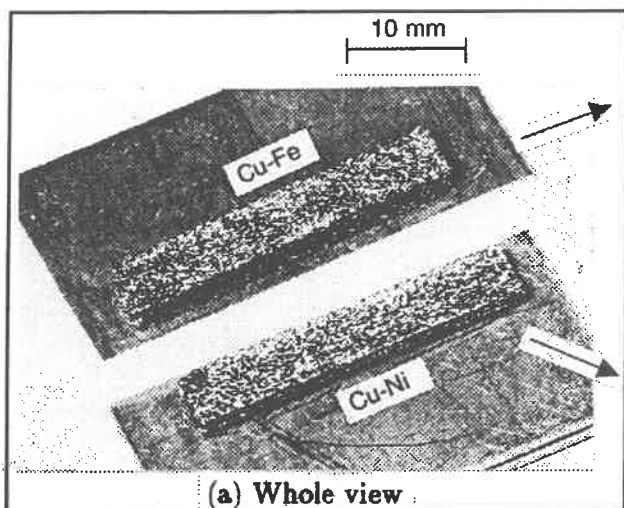
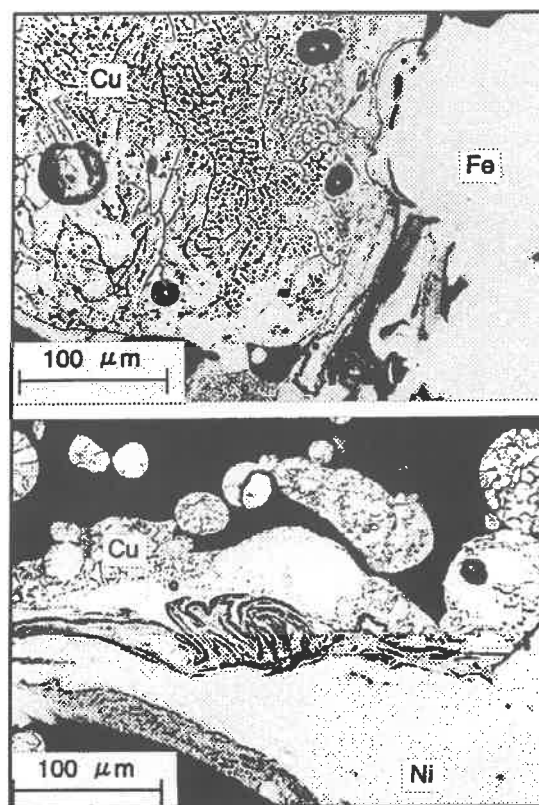


Fig. 4 Hybrid Cu-Fe and Cu-Ni structures fabricated by the proposed method.



(b) Section view

seen. The top view reveals that the sintering mechanism is not diffusion bonding but solidification of a liquid phase. The degree of melting and the network structure formed by hatching are important when building a layered object. The metallographic examination, Fig. 3 (c), shows that crystallization takes place in the agglomerates. Particles smaller than $50\text{ }\mu\text{m}$ are powders that have not been sintered. An average tensile strength of 20 MPa is obtained [3].

So far, both nickel and stainless steel have shown poor bond strength when a layered object has been made under the conditions listed in Table 1. By combining both materials with copper layer by layer, however, we succeeded in forming hybrid objects, shown in Fig. 4: (a) the whole view, and (b) the microstructures photographed by a metallurgical microscope, where the copper only was etched by a corrosive agent. It is found that agglomerated grains coalesce at the boundary to form an alloy. In particular, the boundaries of copper and nickel look autogenous or self-smelting. The black portions of the figures are pores generated during sintering. Closed circular pores stem from vaporization. Further adjustments in sintering conditions and infiltration or sinter casting are necessary to obtain full-density composites. Compared to a simple copper substance, the Cu-Fe system gives a higher density of 5.0 g/cm^3 and a tensile strength of 62 MPa.

Complex copper objects

Using the "green-tape sintering method", we can fabricate objects with various shapes and functions. Figure 5 shows examples made of copper powder: (a) hollow cubes with rectangular holes, (b) a cooling stage, and (c) a unified wheel-shaft artifact. The left-hand cube is as sintered; this is then heated in a furnace to about $400\text{ }^\circ\text{C}$ to resolve the organic binder. The interior powder is then removed to produce the cube on the right. Oxidation darkens the surfaces.

The hollow cube can be adhered to a stainless steel plate. The irradiation power is increased when the first copper layer placed on the substrate is sintered. Therefore, no additional welding

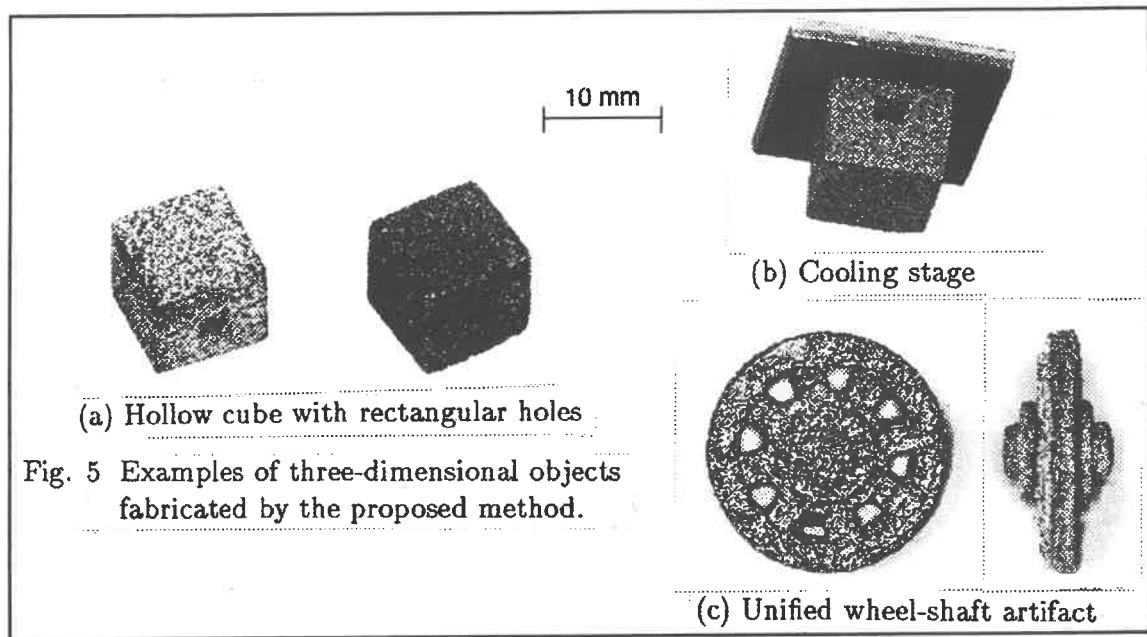


Fig. 5 Examples of three-dimensional objects fabricated by the proposed method.

process is needed. The part may be used as a cooling stage for a deposition device.

The wheel-shaft system is fabricated by the proposed method alone, without any assembly processes. The wheel rotates around the shaft although the dimensional accuracy is not sufficient. The prototype should be improved in accuracy and by further unification.

CONCLUSIONS

- (1) Further developments in laser micro fabrication using powder tape [3] have been demonstrated. Major improvements are twofold: fabrication of hybrid structures by layering more than one green tape, and optimization of sintering conditions which enables us to make complicated objects peculiar to the proposed method.
- (2) The Cu-Fe and Cu-Ni systems have been realized by the partial liquid-phase sintering of micro powders. Tensile strength of the Cu-Fe composite increases roughly threefold, compared to a simple copper substance.
- (3) Complex copper objects have been fabricated by the proposed method, including a cooling stage, a hollow cube with rectangular holes and a unified wheel-shaft artifact.

ACKNOWLEDGMENTS

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Precision Machining of Arrayed Fiber Ends for a Bare Optical Fiber Connector

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1. Introduction

The demand for high-density connection of multiple communication lines to achieve high-speed, large-capacity optical communication systems is increasing recently. The optical bare fiber connector technique is being developed as a solution to satisfy this demand^{1),2)}. In this paper, we describe the development of new precision machining equipment for the arrayed fiber ends of the optical bare fiber connector^{3),4)}. The fiber-end planes need to have a tapered edge for insertion and need to be slightly convex for the physical contact. With this new equipment, we precisely and simply machine arrayed fiber ends at the same time at a reasonable cost.

2. Design of the machining process

To machine the taper, we have developed new machining equipment (Fig. 1), where all the fibers are held perpendicular to a lapping film at the same length (L) from the ends. Then the fiber holder moves in a circular trajectory parallel to the lapping film, but in a fixed orientation to it. If the height between the holding point and the lapping film (h) is shorter than L , all the fibers are bent back against the direction of movement. The bending moments act as a downward force for machining and the contact point at the end plane of the fiber is machined. As the holder moves through its trajectory, the contact point moves around the plane edge of the fiber, and the edge of the fiber end is machined to a taper shape whose angle is determined by the bending angle at the contact point. With this method, since the optical fiber has a constant shape and is highly uniform, the downward force caused by the bending moment is uniform during machining; thus precision taper machining can be done. All the fibers experience the same machining at the same time without interfering with each other if the bending is not excessive. Figure 2 shows a schematic diagram of our machine. The driving mechanism is supported on three micrometers. The holder height can be varied by adjusting these. The taper shape, i. e., the taper

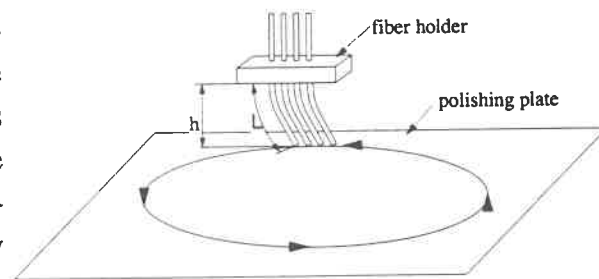


Fig. 1. Principle of taper machining.

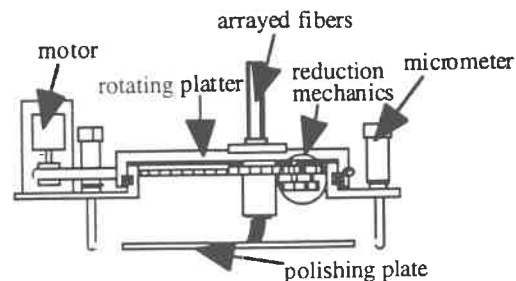


Fig. 2. Schematic diagram of tapering machine.

angle (θ) and the residual plane diameter (d), can be controlled through the holder height (h), the fiber holding length (L), and the number of revolutions. For test specimens, we used groups of 8 to 12 arrayed-fibers with pitch of 250 μm .

To polish the fiber-end planes to a convex shape, we used the same type of equipment. With this equipment we obtained the same relative motion as that provided by a well-designed polishing machine, where the work and polisher have the same rotating speed. The fibers were held near their ends and pushed into the polishing plate at 10 to 40 mN per fiber. The fiber-end planes were machined to be slightly convex by the elastic strain distribution of the polishing plate under the fiber ends. We measured the mean end-plane inclination to the perpendicular of the fiber axis using the setup shown in Fig. 3. Here, the highly precise fiber hole of the optical connector ferrule enables us to measure the precise inclination of the fiber ends. To obtain high optical return losses from the polished end planes, we used a newly developed silica abrasive lapping film that also makes the lapping process easier.

3. Results and discussion

Since the taper shape can be controlled through the L and h parameters and the number of revolutions of the fiber holder, we investigated the change in the shape as these parameters were varied. Figure 4 shows the shapes of machined taper fibers. The residual end planes were well-shaped circles with the same radius. θ is the taper angle, and d is the residual diameter of the end plane. The relationship between the shape and the number of revolutions is shown in Fig. 5, where L was 12 mm and h was 10 mm. A 5- μm -diameter alumina lapping film was used. The residual diameter decreased with the number of revolutions, but the taper angle remained constant. The decrease was not linear because the contact area, and thus the pressure of the downward force changed as the machining continued. Figure 6 shows that the taper angle decreased as the bending amount (L - h) increased for all value of L . The residual diameter showed a dependence on the bending amount (Fig. 7). At a given bending amount, both θ and d increased as L increased. Thus the shape is clearly depend on L , h and

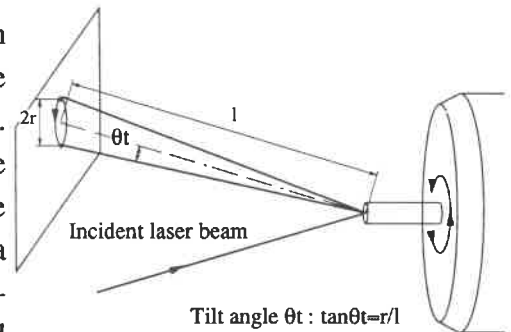


Fig. 3. Tilt angle measurement.

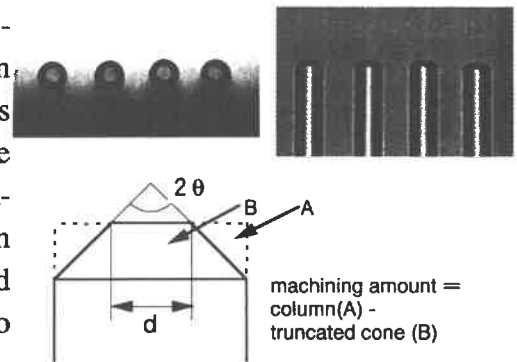


Fig. 4. Machined taper shapes of arrayed fibers. The diameter of the fiber is 125 μm

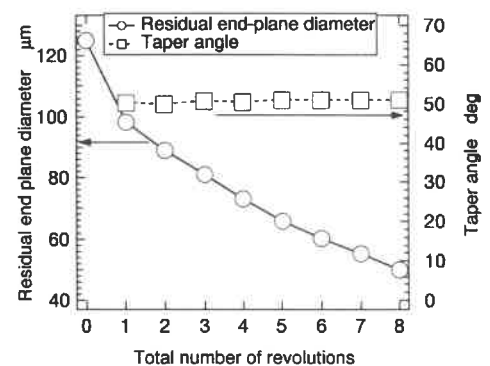


Fig. 5. Change of shape with total numbers of revolutions.

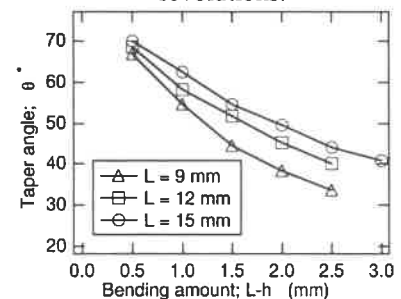


Fig. 6. Change in the taper angle with the bending amount; $L - h$

the number of revolutions.

To understand the underlying principle of this taper machining, we measured the load with an electrical scale placed in the machine instead of the polishing plate. The load increased as the bending amount increased and as L became shorter (Fig. 8). The load was 1 gf per fiber with 2.0 mm of bending amount and L of 12 mm. Figure 9 shows that machining amount calculated as shown Fig. 4 is linearly related to the bending force. Preston's equation shows that the removal rate is proportional to the product of the downward force pressure and the work speed. Substituting this relation, the machining amount is proportional to the product of the load and the work traveling length. It is known that this relation can apply to the lapping region. Figure 9 agrees with this, and indicates that this relationship is the underlying principle of this taper machining.

The fiber-end plane polishing is a two-step process. The first step is to machine the fibers to the same length from the holder with a 3- μm -diameter diamond lapping film. The smooth and almost damage-free planes are then machined with the silica abrasive lapping film in the second step. The slightly convex radius in fiber-end plane polishing needs to be controlled. To do this, the radius change as a function of the polishing load (12 to 42 mN per fiber) and as a function of the fiber length from the holding point (0.2 to 0.6 mm) were investigated. The 3- μm -diameter diamond lapping film was set directly on the stainless plate. The radius decreased as the load increased, although it was constant with the length increase (Fig. 10). The radius changed from about 8 mm to 16 mm which matches the curvature range for physical contact. This suggests that the curvature is determined by the elastic strain distribution of the lapping film near the polished plane. Our next step was to study the influence of the material elasticity on the radius under three conditions: (1) using free diamond abrasives and a fused silica plate as a lapping plate, (2) using a lapping film on the stainless plate, and (3) using silicone

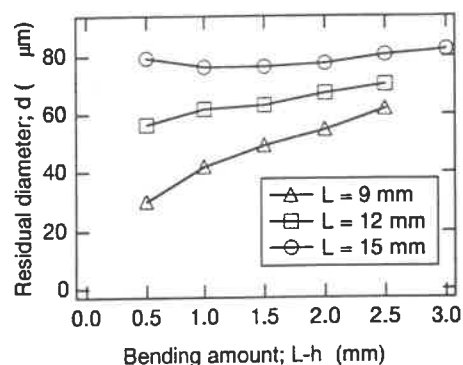


Fig. 7. Change in the residual diameter with the bending amount; $L-h$.

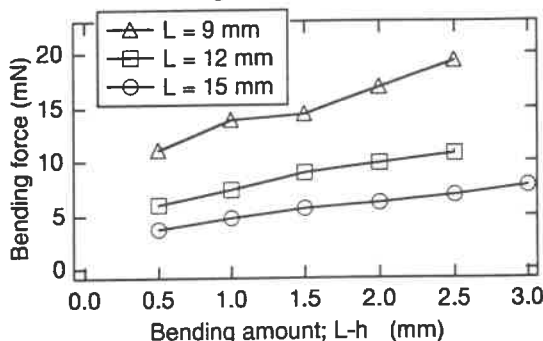


Fig. 8. Bending-force measurement.

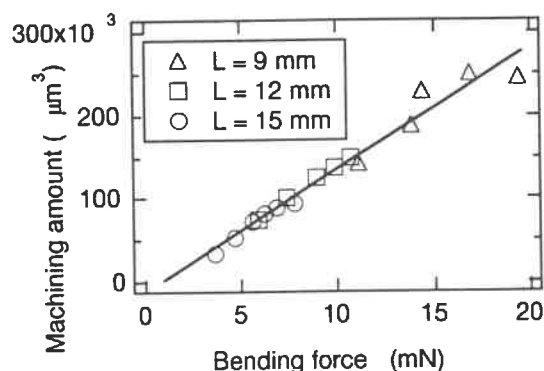


Fig. 9. Relation between machining amount and bending force.

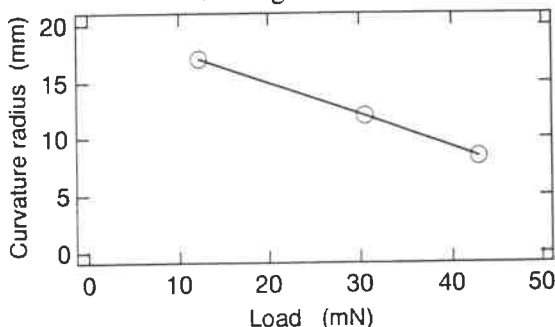


Fig. 10. Curvature radius change with load.

rubber between the lapping film and the stainless plate. The lapping film was a rather hard PET plastic film that is 75 μm thick. The rubber hardness of the 1-mm-thick rubber plate was 50 (ASKER-C). The radius became 60

mm under the first condition, while condition 3 yield on radius that was almost the same as that from condition 2. The calculate height difference between the center and the edge of the 125- μm -diameter fiber-end plane was 0.1 μm for the curvature radius of 10 mm. To achieve this radius the necessary polishing load is only 30 mN. Thus with these results, it can be concluded that the radius is determined by the elastic condition of the thin and rather hard lapping film very near the plane of the polished fiber. Table 1 list the mean plane inclination to the perpendicular of the fiber axis. With precise holder alignment, the inclination was less than 0.5 degrees. Using the new silica abrasive lapping film, we obtained optical return losses of 55 to 60 dB after only 3 minutes of machining.

Table 1. Tilt of polished end planes (deg).

location	1	2	3	4	5	6	7	8	ave.
tilt angle degree	0.3	0.2	0.1	0.1	0.1	0.2	0.1	0.3	0.18

4. Conclusions

We have developed a precise process for machining arrayed optical fiber ends to have tapered corners and slightly convex planes for use as the bare optical fiber connectors. The taper shapes are determined by the length L of the fiber from holder to the fiber ends, the height h of the holder, and number of revolutions. The underlying principle is that the machining amount is proportional to the product of the load and the work traveling length. The curvature radius (8 to 18 mm) of the slightly convex fiber end plane can be controlled by adjusting the polishing load. The polished plane with the silica abrasive lapping film has high optical return loss (55 to 60 dB).

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Quartz-Chip Laboratory for Chemistry, Biochemistry and Quantum Mechanics

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1. Introduction

To analyze micro-phenomena for chemistry, biochemistry and quantum mechanics, a silicon "on-chip laboratory" using MEMS or a carbon "nano-tube" instrument is developed now. On-chip laboratory executes experiments with less sample volume, shorter experiment time and higher measurement resolution.

However, laser lights for various measurements require a transparent test cell even at an ultraviolet region. The authors introduce an on-chip laboratory with a quartz glass substrate. Quartz glass has light transparency even in an ultra-violet region, and chemical stability. Fig. 1 shows a schematic of the quartz-chip laboratory, integrated with some chemical / biochemical apparatus in the surface. It integrates micro reactive / detective chambers, optical devices for observation / measurement and handling tools for pick-up / removal. It also should have many micro fluid-dynamic mechanisms : pumps, mixers, separator, heaters, flow rate sensors, pressure sensors and so on. Apparatus in the surface structure are connected by a network of channels.

In this paper, machining methods to fabricate structures in surface of quartz glass substrate will be described. As applications of quartz-chip laboratory, some prototypes on molecule detection by thermal lens microscope, DNA fiber extraction and quantum mechanics analysis, and some surface-structured apparatus will be shown.

2. Machining method of quartz glass

Sizes of structures in surface area of a glass substrate spread from a sub- μ m range to a mm range. We developed two fabrication processes to the size of the structures. For 10 μ m - 10mm features, powder beam blasting is employed. It blasts alumina powders of 1 - 20 μ m diameter with a lithography tape-mask on the surface. For 10nm - 10 μ m features, fast atom beam (FAB) etching[1] is employed. Mask pattern is so fine that electron beam lithography is used for drawing.

Ion beam etching is used generally to fabricate fine structures. But when the ion beam etching is applied to materials which do not have conductivity like quartz glass, the surface is charged, and ion beam is reflected. Fig.2 shows

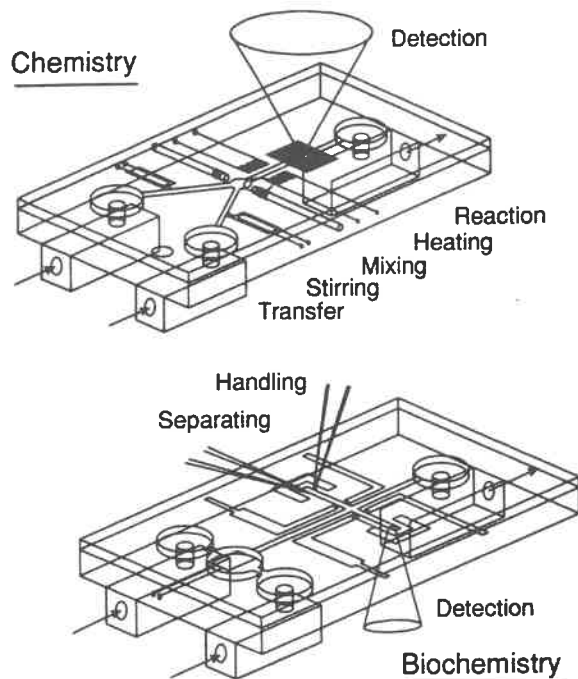


Fig. 1 Schematic view of applications of quartz-chip laboratory

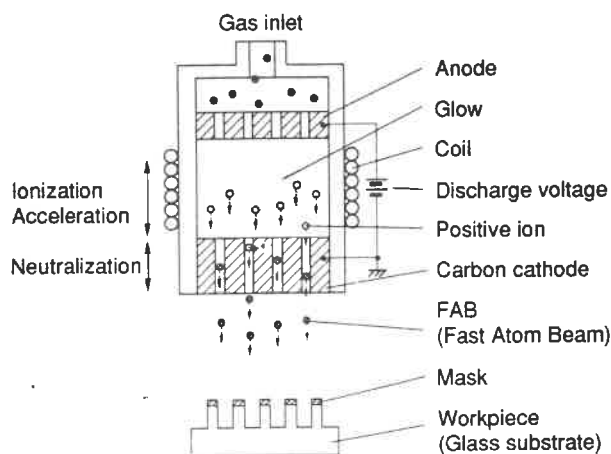


Fig.2 Mechanism of FAB source

a mechanism of the FAB source. Gas is ionized and accelerated by discharge voltage which is applied between anode and cathode. Positive ions are neutralized when they go through the cathode, and neutral fast atom beam (FAB) is obtained. As the FAB is neutral, it is a suitable machining method for glass substrate.

3. Applications of quartz-chip laboratory

3.1 Thermal lens Microscope

Prof. Sawada and Kitamori proposed molecule detection by a "thermal lens microscope" [2] [3]. The thermal lens microscope has detection capability of several molecules and can be used for quantitative / qualitative measurements. We developed a prototype of quartz-chip laboratory for the thermal lens microscope.

Fig. 3 illustrates principle of the thermal lens microscope. An excitation laser beam is absorbed in microparticles. The absorption causes change of temperature of surrounding solution. The temperature change causes change in refractive index of the solution. This area acts as a concave lens and changes curvature of a probe beam. The curvature change induces an intensity change of the probe beam detected through a pin hole. It can detect microparticles in a liquid or a fluid with an extremely high resolution of several molecules.

However, turbulence of solution flow causes double-count measurement error of microparticles. In an extreme case, thermal Brownian movement of the particles may be a cause. A narrow lamina-flow channel called "nano-channel" is employed to increase preciseness of the measurement.

Fig. 4 indicates a nano-channel unit designed for measurement area of the thermal lens microscope. Several channels run parallel. The channels have dimensions of 900 nm width, 20 μ m pitch, 1 μ m depth and 1 mm length. Laser beam of the thermal lens microscope select one channel for detection. To decrease fluid friction on the channel except the measurement area, much wider channels are fabricated. Fig. 5 shows a whole view of a fabricated nano-channel unit and its magnifications.

The nano-channel unit is covered with a thin glass plate. Porphyrin in benzen solvent is employed as a sample solution, and the channels are filled with the solution. Density of the solution is adjusted so that 100 molecules of porphyrin exist in the detection area of the thermal lens microscope. When the laser beams of the thermal lens microscope scan across the channels, signal peaks appear at the same pitch as that of channels. Now we are trying to detect molecules in lamina-flow in nano-channels.

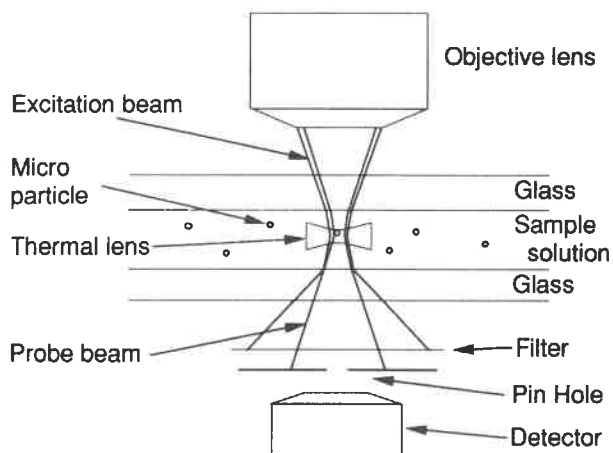


Fig.3 Thermal lens microscope

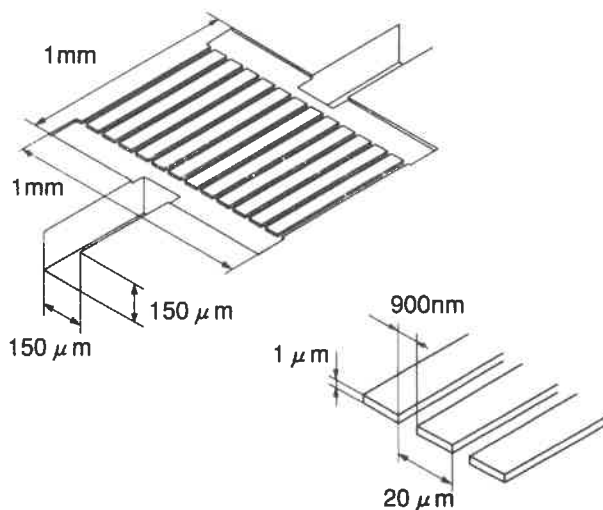


Fig.4 Structure of nano-channel unit

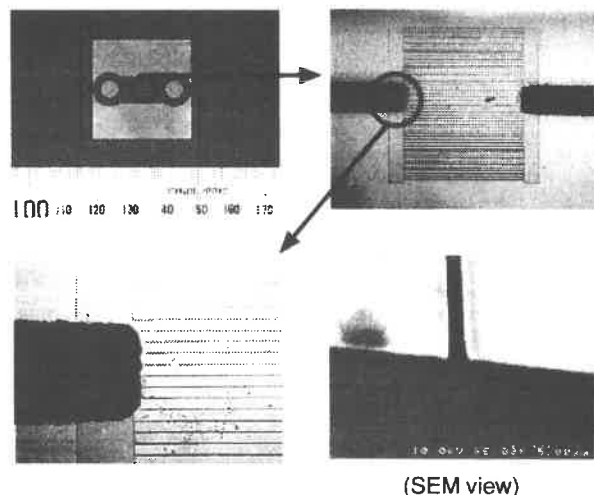


Fig. 5 Fabricated nano-channel unit

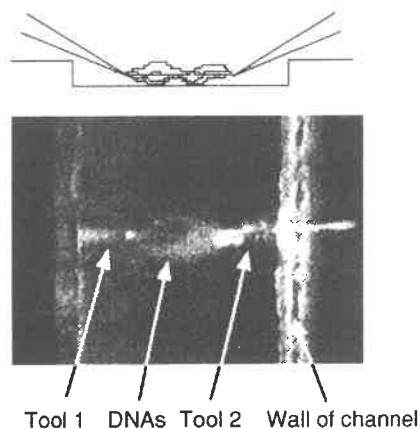


Fig. 6 Optical view of picking up DNAs between the tools

3.2 DNA fiber extraction

DNA "surgery" is one of the hottest topics these days. To extract a DNA chain from a single chromosome is a highly desired technique. To achieve the extraction, a prototype of quartz-chip laboratory are being developed.

Prof. Washizu has demonstrated stretch-and-positioning of DNA by electrostatic field[4]. We demonstrated DNA handling by electrostatic field applied to needle tools. Fig. 6 shows two tools picking up DNAs. The tools are glass tubes of $0.8 \mu\text{m}$ diameter, sputtered aluminum thin film outside and filled with luminescent liquid inside. Electrostatic field is about $1 \text{ V}/\mu\text{m}$ at 1 MHz .

Fig. 7 shows a schematic of DNA fiber extraction system on a quartz-chip laboratory. Electrodes are formed along a channel. By applying traveling electrostatic field, DNA fiber will be extracted and conveyed.

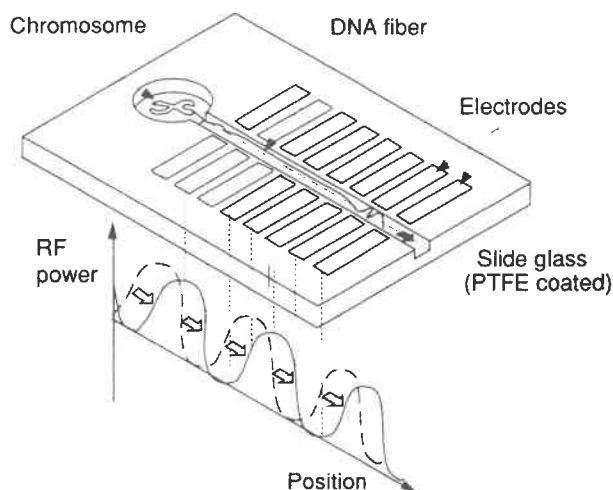


Fig. 7 Schematic of DNA fiber extraction from a single chromosome

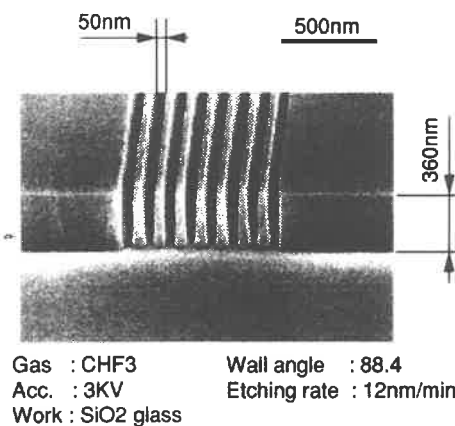


Fig. 8 SEM view of 50nm nano-channels

3.3 Quantum mechanics

Characteristics of liquid confined in nm-scale area are supposed to be quite different from that in normal circumstances. To confirm this, a prototype of quartz-chip laboratory with very narrow channels are developed.

Fig. 8 shows an SEM view of nano-channels fabricated in surface of quartz glass substrate by the FAB. The substrate is etched by CHF3 beam accelerated at 3kV. The channels have 50 nm width, 100 nm pitch and 360 nm depth. The walls of the side and bottom of the grooves have enough smoothness. Now we are trying to measure the characteristics of flow in these channels.

4. Surface-structured apparatus

Some surface-structured fluid-dynamic mechanisms are being developed.

Fig. 9 shows a schematic of a ball stirrer. When two flu-

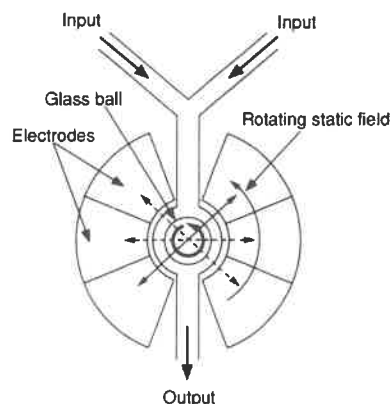


Fig. 9 Schematic of ball stirrer

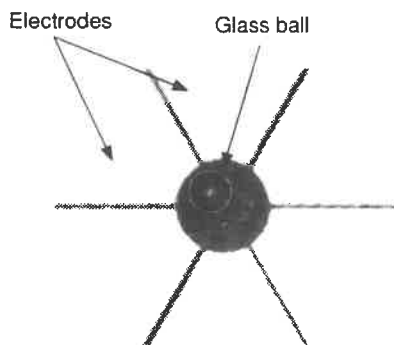


Fig. 10 Photograph of ball stirrer

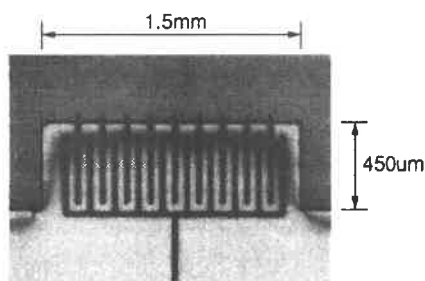


Fig. 11 Photograph of thin film heater

ids are mixed in "Y-letter" channel, they are mixed by their slow diffusion, not by quick turbulence like a violent river. Prof. Washizu proposed micro motor driven by electrostatic field[5]. We applied this method for the ball stirrer. The glass ball is located in the channel, and is surrounded by radiated electrodes. Rotating electrostatic field is applied to the ball, and the ball is rotated.

Fig. 10 shows a prototype of the stirrer. Diameter of a ball is $200\ \mu\text{m}$. 3 couples of electrodes are fabricated with Al thin film. Voltage to generate electrostatic field is $\pm 10\text{V}$. When the field rotates at about 640kHz , the ball rotates at 0.5Hz .

Heater is indispensable to promote chemical reaction. Fig 11 shows a thin film heater for quartz-chip laboratory. The thin film is made of vapor-deposited Cr, and fabricated by an UV laser cutter. Thickness of the glass substrate is 1.05mm , and thickness of the Cr film is 100nm . Resistance of the heater is measured to be $40\text{k}\ \Omega$. Temperature rises to 100°C when input power is 2.3mW . The

temperature is determined by balance of generation and diffusion of heat.

5. Conclusion

On-chip laboratory with a quartz glass substrate is proposed. Quartz glass has light transparency even in an ultra-violet region, and chemical stability. It enables various measurement methods with laser beams.

Machining methods for quartz glass substrate are developed. The FAB etching has machining capability of 50nm wide channels.

As applications of quartz-chip laboratory, prototypes on molecule detection by thermal lens microscope, DNA fiber extension and quantum mechanics analysis are proposed.

Some surface-structured apparatus for quartz-chip laboratory are demonstrated.

Quartz-chip laboratory, even though it is still developed at an early stage, will become a promising science instrument.

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COMPACT PORTABLE AIR JET GUN FOR THERMAL CUTTING OF PLATE GLASS

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1.0 INTRODUCTION

The fabrication of optical lenses and orthodontal photographic mirror blanks involves different stages which include blank preparation, grinding, polishing etc. In this conventional process of blank preparation the number of operations involved are more and there is a possibility of developing internal flaws and minute cracks, which reduce the performance of the lens and mirror [1,2].

In an earlier investigation by the above authors, a new method was developed for cutting the lenses thermally [3,4]. The method of thermal cutting of plate glass consists of propagation of a crack in glass, owing to thermal stresses produced with the help of a moving hot air jet. A crack if already present in the glass, can be extended. The movement of the jet governs the direction of propagation of the crack and thus it is possible to cut a glass plate accurately to any desired size and profile in a single step. Movement of the jet in the desired path can be achieved by using a X-Y table or a pantograph mechanism. A schematic diagram of crack propagation is shown in Fig. 1. This thermal cutting process eliminates processes like chipping and grinding to size the blanks.

Continuing the above work, a compact portable air jet gun similar to hot air blower has been designed and fabricated.

2.0 CONSTRUCTION OF AIR JET GUN

A schematic diagram of the gun is shown in Fig. 2. It consists of two concentric sheet metal tubes. Both internal and external surfaces of the inner tube are clad with a layer of asbestos sheets to prevent loss of heat. A heating coil of 2 kW rating mounted on an asbestos thick sheet is inserted into the inner tube which is brazed to the lower end cap. The lower end cap has a built-in nozzle of 2 mm diameter. The inner tube is introduced into a outer tube. The clearance between the two tubes is filled with glass wool to prevent heat loss.

The upper end cap is provided with an inlet for clean compressed air. A provision is made for fixing fan and electric motor so that the gun acts as a blower.

3.0 EXPERIMENTAL PROCEDURE

When the cut is to be made, an initial crack is obtained about 2 mm in length by a diamond cutter. It should be ensured that air jet is kept in fixed position by properly positioning the glass tube. Then compressed air is injected through the nozzle. A gap of about 2 mm is provided between the glass plate to be cut and the nozzle point. Then voltage is applied to the heating element. Temperature of the air jet can be controlled with the help of the dimmerstat. Then voltage is applied to the heating element. Temperature of the air jet can be controlled with the help of the dimmerstat. The hot air jet, first impinges on the air of the initial scratch. After about 10 seconds, the crack begins to propagate in a straight line, when the glass plate is moved in longitudinal direction with respect to the nozzle.

Having made the initial crack, the crack propagation can be achieved in two ways:

By keeping the gun stationary and moving the glass plate (CNC X-Y table may be used)
By keeping the glass plate stationary and moving the gun (pantograph mechanism may be used)

4.0 RESULTS AND DISCUSSION

Experiments were conducted to cut plate glass (window glass) of 2,4 and 5 mm thickness. Straight cuts were made using a manual X-Y table. Fig. 3 shows the experimental results. It indicates that the cutting time is proportional to the thickness of plate. It was observed that the propagation of crack initially and at the end of the cut (i.e. near the edge of cut) takes more time.

5.0 CONCLUSION

A compact portable air jet gun for thermal cutting of plate glass is designed and fabricated. Experiments are conducted with the gun to cut glass plates of different thickness. Results showed that the cutting time increases with increase in plate thickness. This compact gun can be used for precise cutting of optical lense blanks and ortho-dental photographic mirrors.

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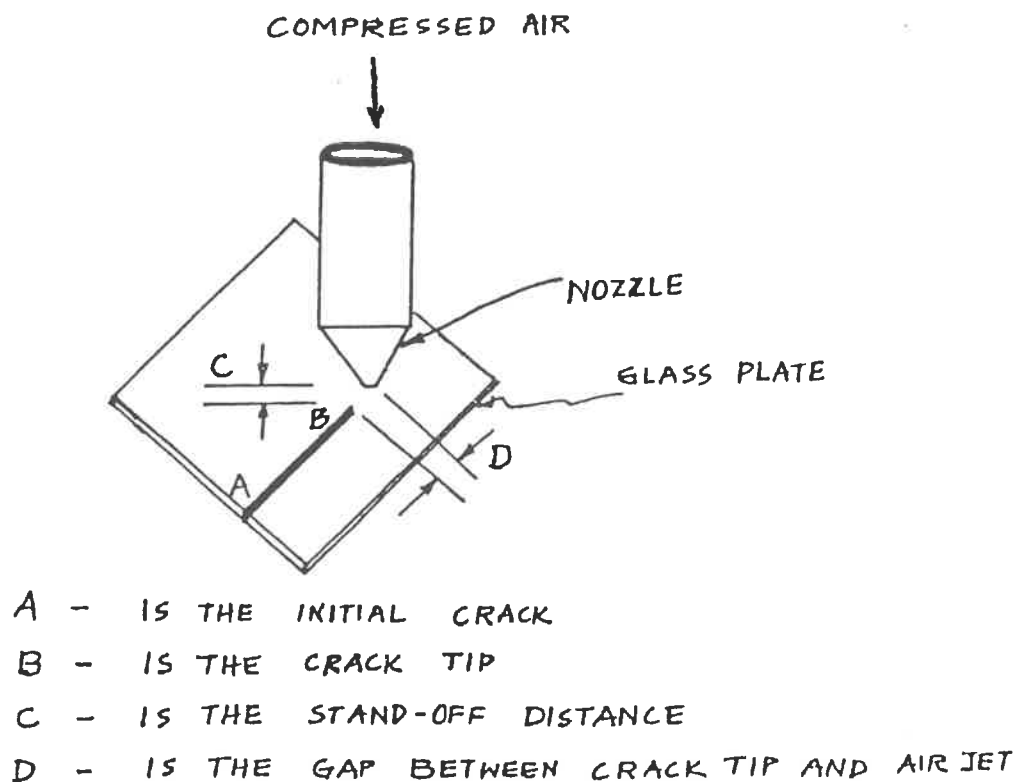


Fig. 1 Crack propagation

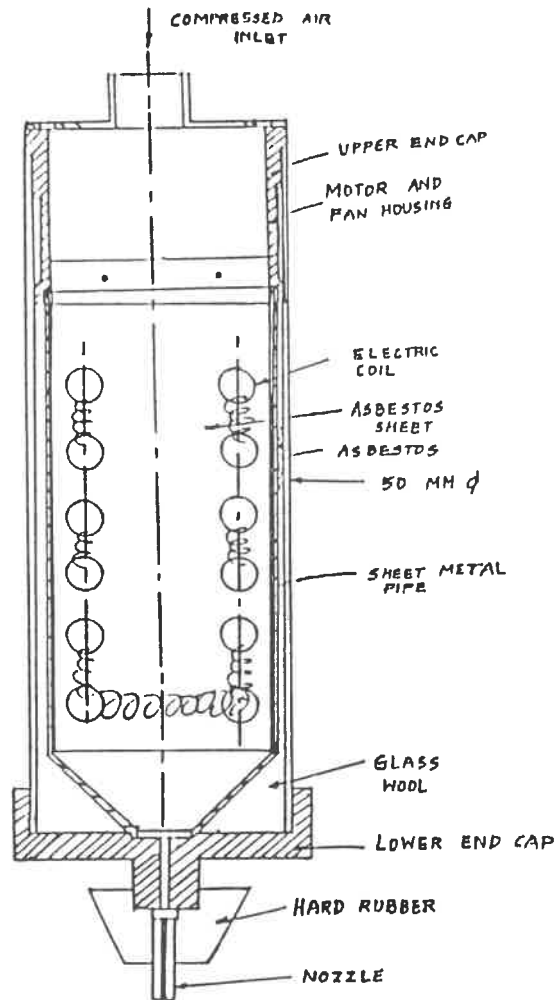


Fig. 2 Compact hot air jet gun

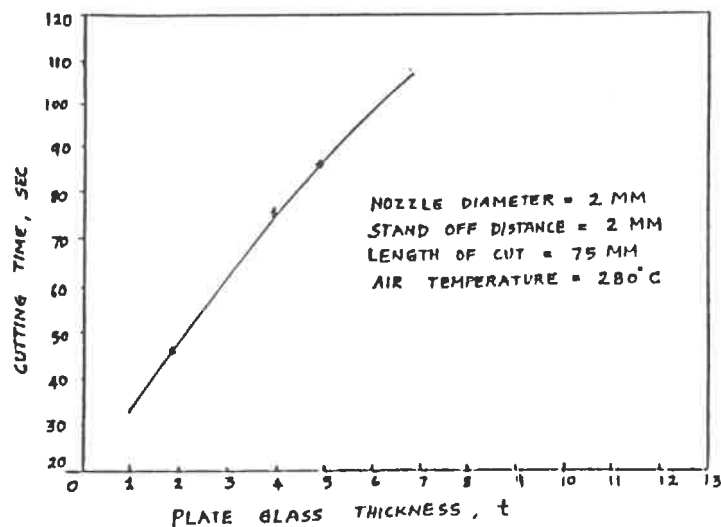


Fig. 3 Variation of cutting time with plate glass thickness

Development of a Shape Generation System Using the Layered Fabrication Method

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1. Introduction

Although the CAD system is widely utilized in shape design, many hours are required to use all system functions to produce a shape matching the intentions of a designer. In particular, general designers have difficulty using higher mathematics containing partial differential equations, such as Bezier curves, and therefore have trouble managing changing systems for complicated processors involving 3-dimensional sculptured surfaces. This research therefore proposes a system for generating shapes more easily using the layered fabrication method. With this method a relatively new CAD system user can generate shapes rather easily. The method is also applicable in the learning and manufacturing areas. A rapid prototyping system¹⁾ using paper or thermosetting resin already exists for actual shape generation using the layered fabrication system. However, a feature of the proposed system is the ability to generate complete 3-dimensional shapes using cross-sectional layers combined with computer simulations. An outline of the proposed system is provided in this report along with an explanation of cross-section generation and the layered fabrication method. Some simple examples are also described.

2. System Outline

The basic method of the system is shown in Fig. 1. The contours of shapes are first extracted from painted in diagrams and cross-sectional shapes are generated. Next, the cross-sections are layered to create a 3-dimensional shape. As shown in Fig. 2, the general procedures of the system involve 2 steps. In the first step cross-sectional shapes are generated and in the second step cross-sections are layered to generate a 3-dimensional shape. In cross-sectional shape generation, the contour points of a cross-section are first determined. The contour points are then connected to create the final cross-sectional shape. In 3-dimensional shape generation, the base points of cross-sections are determined and a 3-dimensional shape is generated by layering. A more detailed explanation follows.

3. Cross-sectional Shape Generation

As shown in Fig. 3, a cross-sectional shape is generated in an $M \times N$ lattice set in the X-Y plane. The designer uses a mouse to paint a desired shape into the lattice. The external contour

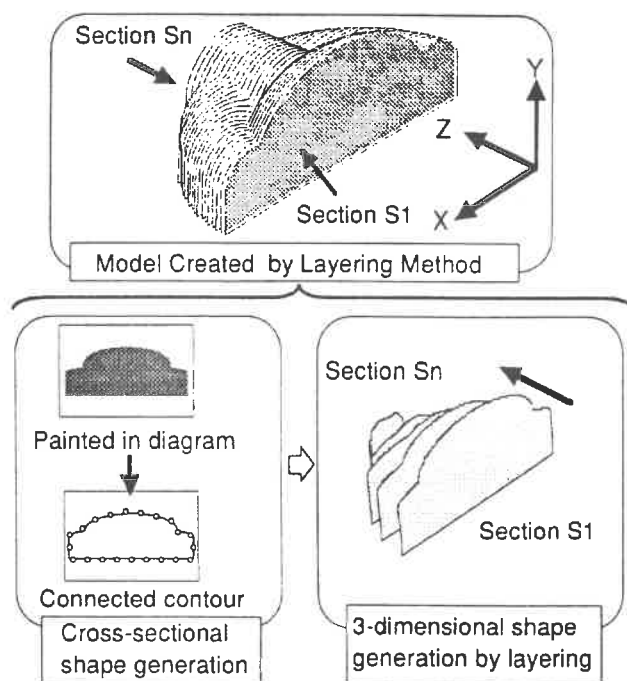


Figure 1 Model Shape Generated by Layering Method

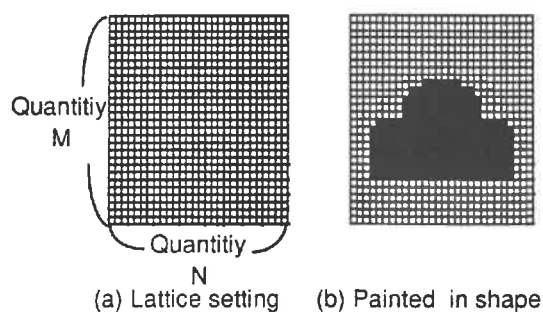


Figure 3 Cross-sectional Shape Defined on Lattice

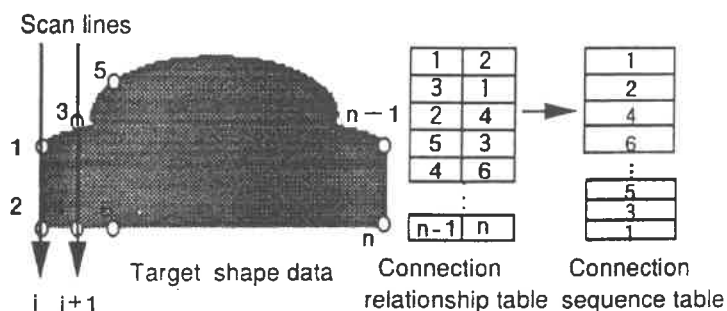


Figure 4 Contour Extraction for Painted In Shape

Next, any point can be used as the start for the connection sequence. For example, when 1 is used as the starting point, the paired point (2) seeks another point with a connection relationship with

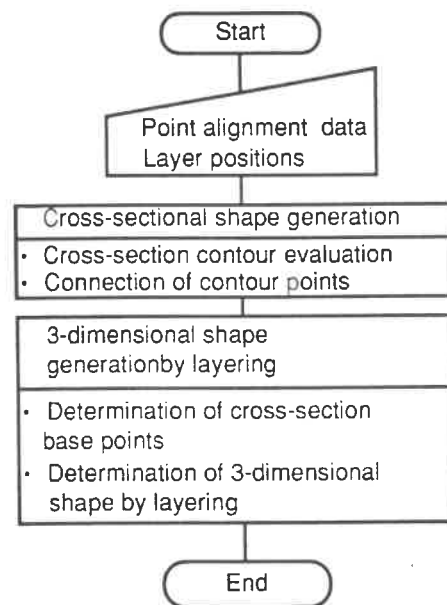


Figure 2 General Processing Outline for Shape Generation System

lines are then extracted, using the following connection algorithm²⁾, from this painted in shape. Fig. 4 shows the method for extracting contour lines from painted in shape data. While scanning a shape using 2 lines, the contour section is evaluated and connected. In other words, the scan lines of points i and $i+1$ are taken together and pairs are created starting from the lowest Y-coordinate value. When the y-coordinate values are the same, pairs are created from the lowest X-coordinate value. These contour point pairs are arranged in a connection relationship table.

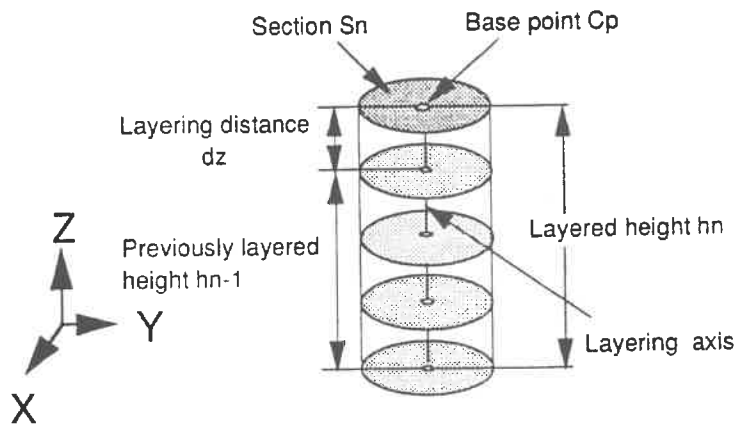


Figure 5 Shape Creation by Layering

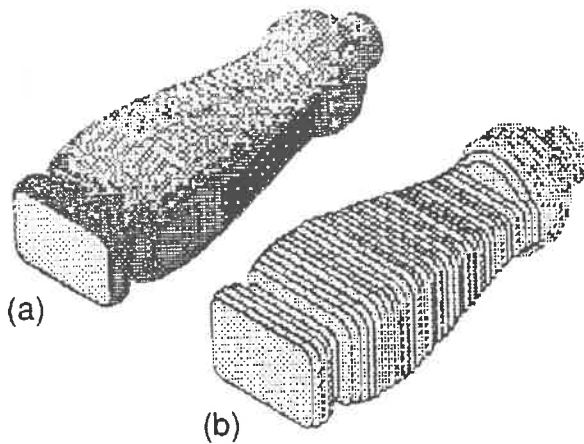


Figure 6 Comparison between Conventional Method and Layered Method

itself. The point at 4 is obtained as a result. The connection sequence table is created by repeating this process. In this way the contour shape can be extracted.

4. Shape Generation by Layering

Fig. 5 shows the method for generating 3-dimensional shapes through layering. This method simply involves layering cross-sectional shapes from bottom to top. In other words, when a designer sets base point C_p in cross-section S_n , which is currently being added, the position of the cross-section in the X-Y plane is determined. The new layered height, h_n , is also set at the same time. The sum of the previously layered height, h_{n-1} , and the layer depth, dz , is the new

h_n . The above processing is repeated for all sections to generate the final 3-dimensional shape.

5. Examples

The above processing was programmed into a computer and the model shapes shown in Fig. 6 were generated. The shape in Fig. 6(a) was generated using a conventional CAD system and the shape in Fig. 6(b) was generated using the layering method proposed in this report. As can be seen, the shape in Fig. 6(a) is approached more closely as the distance between each layer is reduced. In addition, 3-dimensional shaping time can be reduced by reducing the distance between layers where the change is great and increasing the distance between layers where the change is small³⁾. This example shape was generated using the personal computer. When the section thickness was 1mm and the distance between sections was 2.5mm, the shape could be generated in less than 1 hour.

6. Conclusion

In order to simplify shape input in design processes, a prototype system was created for generating shapes by layering cross-sections created from point data. The following results were thus obtained.

- (1) Through simple shape input using point data, intended shapes could be generated without specialized knowledge.
- (2) While cross-sectional shapes are layered, the final shape can constantly be checked on the display screen.

In the future, through addition of sweep functions and cross-sectional shape expansion and contraction, a more generalized shape generation system will be realized.

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Formation of Rainbow Patterns on Die and Mold Surfaces by Micro-Indentation Method

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Abstract: This paper proposes a new method for forming rainbow patterns directly on dies and molds by a piezoelectric vibration assisted micro indentation method in the aim to realize high value-added injection molded plastic products. In experiments, regular micro-pits were formed by indenting a diamond indenter with a conical tip against the mold surface. The experimental results showed that when the intervals at which the indenter is indented are 5 to 7 μm and the depth of the pit formed is below 7 μm , the rainbow pattern can be formed clearly. The formation of the rainbow pattern was found to be unaffected even when the depth of cut and indenter tip radius were changed. Experiments on injection molding using a mold with rainbow characters and patterns also confirmed that the rainbow pattern can successfully be transferred to plastic molded products.

Key words: Rainbow pattern, Injection molding, Die and mold, Vibration assisted micro indentation method, Piezoelectric actuator

1. Introduction

The tasks currently faced in injection molding include (1)high efficiency molding, (2)high precision molding, (3)molding of difficult-to-machine resins, (4)molding of complicated shapes, (5)molding of micro parts, (6)improving recycling performance, (7)enhancing added-value of products by adding designs, etc. To deal with these tasks, studies are being carried out from various aspects such as injection molding device, die and mold, molding conditions, flow analysis, material, cavity surface processing, etc. The aim of this study is to build a mold for forming a rainbow pattern on one part or over the whole surface of plastic molded products as a means to realize (7) high value-added plastic molded products by adding designs.

2. Study of Formation of Rainbow Design on Die and Mold

Since a rainbow pattern is produced by the diffraction phenomenon of light, a rainbow pattern can be formed on a mold by forming micro dents on the cavity surface of the mold. The dents can be in the form of linear grooves or pit-shaped as demonstrated by diffraction gratings and compact discs, etc. Methods for forming the dents on metal materials include (1)cutting, (2)etching, (3)shot-blasting, (4)laser cutting, (5)plastic indentation, etc. However the (5)plastic indentation method is considered the best method taking into consideration the restriction of the pitch of the dents under 7 μm [1] to form the rainbow pattern as well as the need to form complicated rainbow characters and patterns.

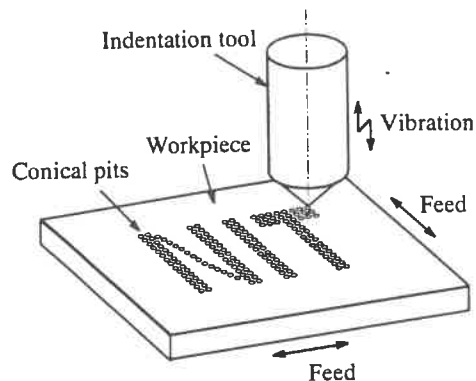


Fig.1 Schematic illustration of micro-indentation method for forming rainbow pattern by piezo-transducer-device

In order to form the enormous number of micro pits composing the rainbow pattern efficiently, the authors propose a method of mounting a hard conical or ridge-shaped tool to the tip of an ultrasonic vibrator or piezoelectric vibrator and striking the mold with this tool (micro indentation, Fig.1).

3. Equipment for Formation of Rainbow Pattern

To form desired rainbow patterns on a mold surface efficiently, the indenter is pressed at high speed while moving the work mounted on the NC machine table. Although an ultrasonic vibrator can be used as the source for driving the indenter to and fro at high speed, in this experiment, a piezoelectric actuator which can

control the pit pitch to a uniform level according to the change in the table feedrate was used.

4. Experiments on Formation of Rainbow Pattern

4.1 Indentation Device

Figure 2 shows the schematic construction of the piezoelectric transducer device for forming rainbow patterns. The vibrator used in the experiments is a piezoelectric device (ASB170C801NP0, Tokin) sealed in a pre-pressurized metal case. The diamond indenter built inside the fixed jig and fixed to the tip of the vibrator is indented by vibration. The frequency can be varied by the function generator (to 1 kHz). An air-cooled hole is also provided to control the heat generation of the vibrator.

4.2 Experimental Conditions and Methods

In this experiment, an attachment incorporating the piezoelectric transducer device was mounted to a longitudinal MC spindle. The frequency of the device was set at $f=500$ Hz, the amplitude was set at $\delta=10 \mu m^{p-p}$, and square vibration waveform was used. To form the rainbow pattern, a ridge-shape diamond indenter ($\phi 3.1$ mm, peak angle= 130° , radius= $8, 110 \mu m$, **Fig.3**) was indented on a NAK80 ($H_R C40$) mold steel surface that had been lap-finished ($<0.007 \mu m Ra$). **Figure 4** shows the external view of the experimental device, while **Table 1** shows the other experimental conditions.

4.3 Evaluation Items

The rainbow pattern is required to satisfy the following conditions; (1)good shine (high brightness), (2)good spectrum (divides into red, blue, green, etc.), (3)extensive angle of view, (4)various light source conditions (position, type), etc. To satisfy these rainbow pattern features, the indenter conditions (angle, radius of tip) and machining conditions (pitch, depth of indentation) must be set to the optimum values.

Table 2 shows the evaluation items for the rainbow pattern and the influential factors.

In the experiments, the conditions for achieving the optimum rainbow pattern formation were studied.

5. Experimental Results

(1) Effects of pit pitch

Figure 5 shows the results of SEM observations of the rainbow pattern formed when an indenter with a tip radius of $8 \mu m$ was used, and the indentation depth (a) and pit pitch (d) were changed.

Excellent rainbow patterns were formed when the pit pitch was $5 \mu m$ and $7 \mu m$ (visually confirmed). When pit pitch was $10 \mu m$, the rainbow pattern became whiter with the center portion showing the whitish

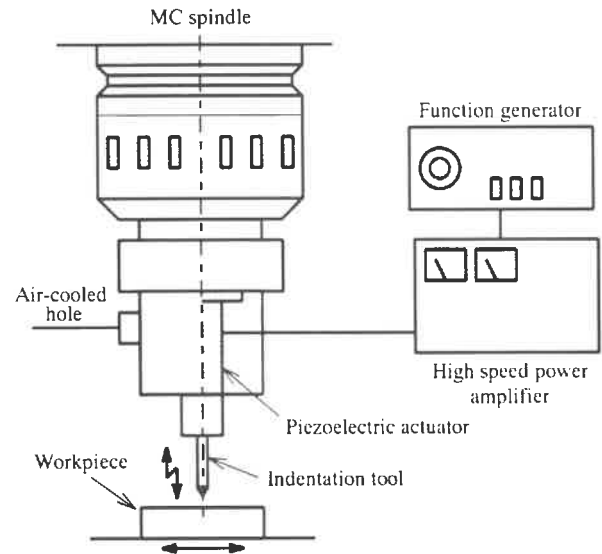


Fig.2 Schematic construction of indentation device

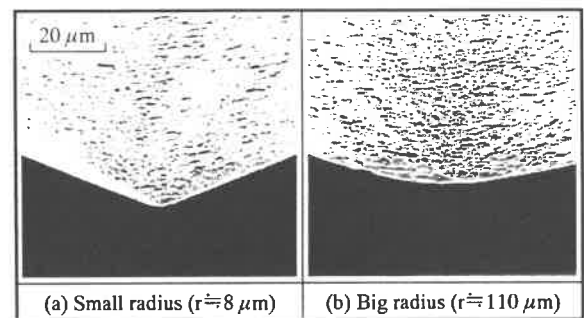


Fig.3 Enlarged view of diamond tool tip
($\theta = 130^\circ$, Conical type tool)

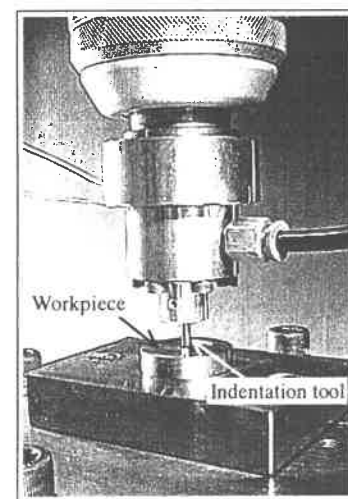


Fig.4 Formation of rainbow pattern by micro indentations with piezo-transducer-device on an MC

color of the background. The rainbow pattern was more or less invisible at $15\text{ }\mu\text{m}$.

(2) Effects of overlapped indentation

The rainbow pattern was successfully formed when the depth of indentation (a) was below $7\text{ }\mu\text{m}$. It was, however, found that, in case of large depth such as $7\text{ }\mu\text{m}$ and small pit pitches such as $5\text{ }\mu\text{m}$, the pit shape formed might collapsed severely due to subsequent indentation.

(3) Effects of indenter tip radius

Figure 6 shows the results of the experiment on the formation of a rainbow pattern using a tool with a tip radius of $110\text{ }\mu\text{m}$. The range of formation of rainbow pattern was found to be more or less the same as an indenter with a sharp tip. The swelling at both ends of the pattern undesirable during injection molding was also found to become greater.

Nevertheless, the use of an indenter with a considerably large tip radius is considered advantageous for the prevention of chipping and wear.

Table 1 Experimental devices and working conditions

Experimental machine	Vertical machining center (MT-V10, Yamazaki Mazak)
Piezo transducer device	Piezoelectric actuator (ASB170C801FP0, Tokin) Function generator (FG-122, NF Electronic Instruments) High speed power amplifier (4025, NF Electronic Instruments)
Indentation tool	Conical type tool ($\phi 3.1\text{ mm}$, peak angle= 130° , Radius= $8, 110\text{ }\mu\text{m}$, Diamond)
Workpiece materials	NAK80 ($H_R C40$, Initial surface : Mirror finished $<0.007\text{ }\mu\text{m Ra}$)
Working conditions	Oscillation frequency : $f=500\text{ Hz}$, Amplitude : $\delta=10\text{ }\mu\text{m}^{PP}$ Feedrate : $V_w=150\sim 450\text{ mm/min}$ (pitch= $5\sim 15\text{ }\mu\text{m}$) Depth of indentation : $a=3, 5, 7\text{ }\mu\text{m}$

Table 2 Evaluation items and influential factors of rainbow pattern

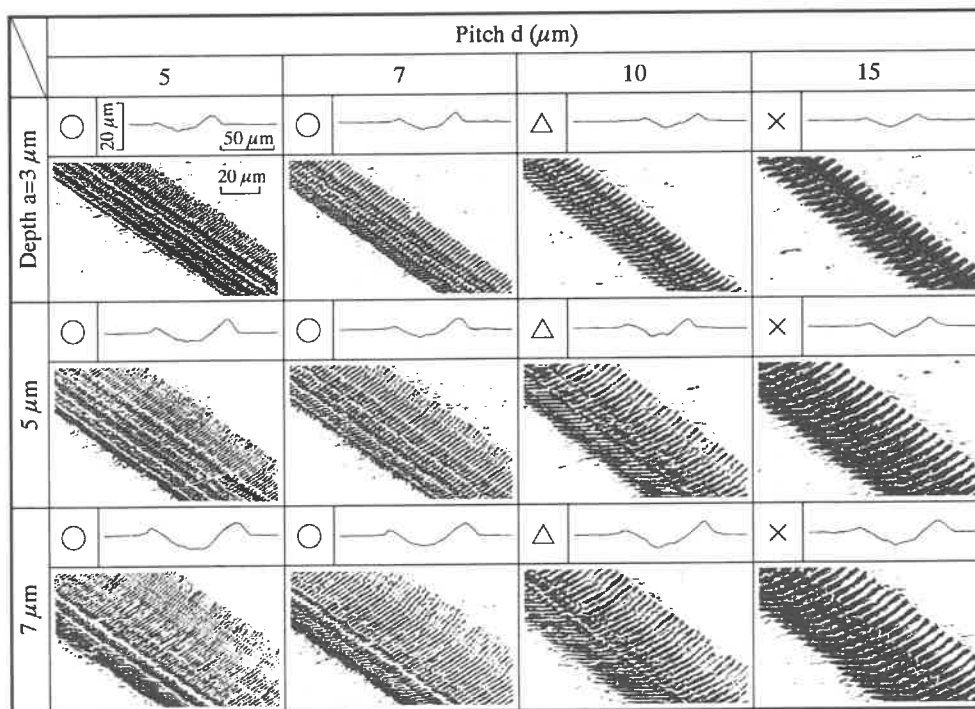
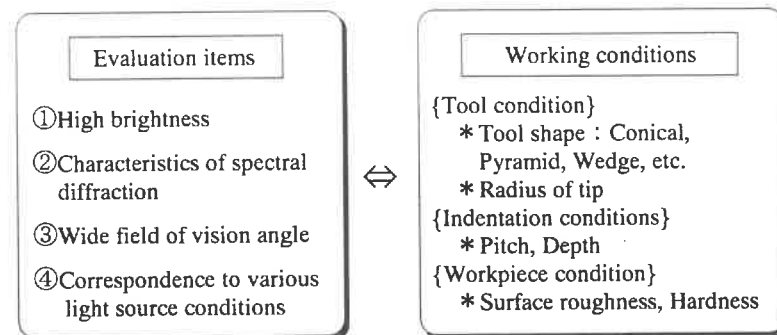


Fig.5 Good and bad rainbow patterns formed by small radius tool ($r \approx 8\text{ }\mu\text{m}$)
($f=500\text{ Hz}$, $\delta=10\text{ }\mu\text{m}^{PP}$, O:Good, Δ :Visible, X:Bad)

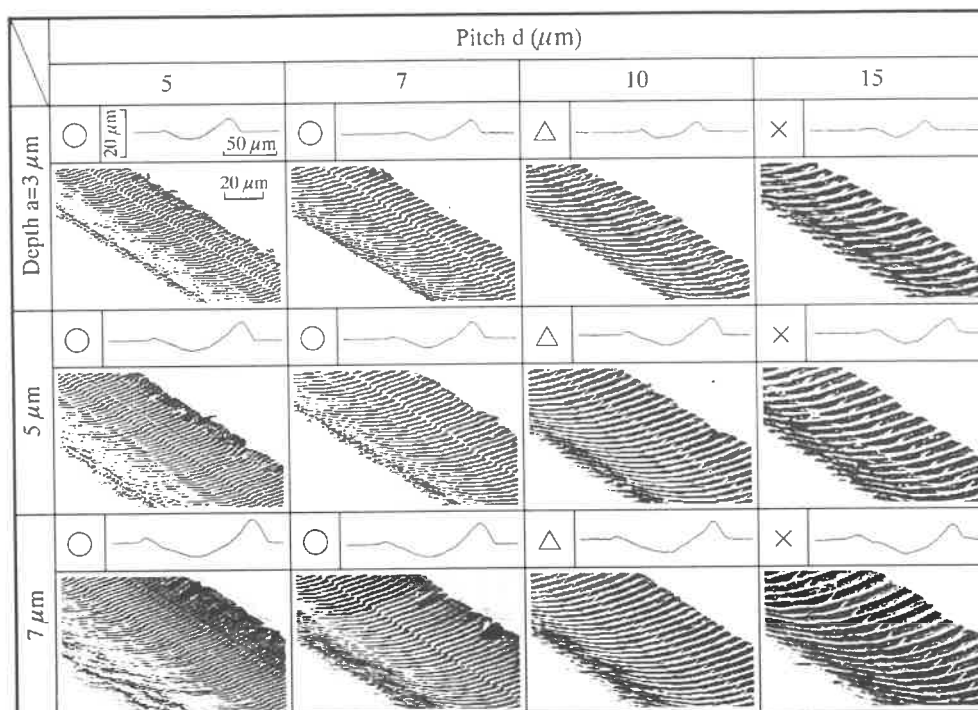
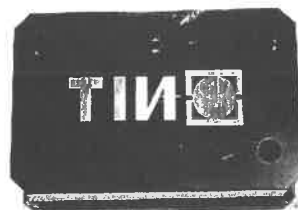


Fig.6 Good and bad rainbow patterns formed by big radius tool ($r=110\ \mu\text{m}$)
($f=500\ \text{Hz}$, $\delta=10\ \mu\text{m}^{\text{P-P}}$, ○:good, △:visible, ×:bad)

6. Tentative Injection Molded Part

The rainbow characters and pattern shown in Fig.7a were formed on a mold surface, and injection molding was carried out using this mold. Figure 7b shows the injection molded part. The plastic material used was PMMA (acrylic resin), and the molding conditions were $T_p=240^\circ\text{C}$, $T_M=80^\circ\text{C}$, and $P=106\ \text{MPa}$.

The rainbow pattern was successfully transferred to the part.



(a) Mold with rainbow pattern formed by micro indentation method
($a=3\ \mu\text{m}$, $d=3\ \mu\text{m}$, $pt=50\ \mu\text{m}$, Conical type tool, NAK80)



(b) Injection molded part with rainbow pattern (PMMA)

Fig.7 Transfer of pattern to injection molded part

7. Conclusions

In this paper, a micro indentation method has been proposed aiming to realize high value-added injection molded plastic products. Based on the results of experiments conducted, the following conclusions were reached.

- (1) The rainbow pattern can be formed by the micro indentation method if the indentation interval is below $d=7\ \mu\text{m}$.
- (2) The rainbow pattern formed is not affected even if the indenter is indented by overlapping.
- (3) The rainbow pattern could be formed with the use of an indenter with a large tip radius, though the swelling at both sides of the pattern increased. The indentation with a large tip radius is considered to be advantageous for the prevention of wear.

- (4) It was confirmed that rainbow characters and patterns can be formed on mold surfaces and that these patterns can also be successfully transferred to tentative injection molded plastic parts.

In pursuing the study, the authors would like to express their heartfelt thanks to Yamazaki Mazak Corp., Nippon Engis Corp., Tokin Corp., and Noritake Diamond Industries Co.Ltd. for their kind assistances.

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A New Micro EDM Technique for Fine Complicated Hole with Triangular Section Electrode

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1. Introduction

In electrical discharge machining(EDM) process, material removal is performed by repetitious generation of a micro crater produced by a single discharge, and the machining force acting on tool electrode and workpiece can be reduced by controlling electrical discharge energy. Therefore, EDM is particularly effective method in micro machining. And then, die-sinking EDM and wire EDM have been widely applied for fine complicated shape hole such as spinneret for synthetic fiber or fuel injection nozzle for engine and so on^{1), 2)}. In these cases, die-sinking EDM has been used when the shape of electrode can be easily made, or otherwise, wire EDM when the forming of electrode is difficult. In die-sinking EDM process, however, it becomes more difficult to make a fine complicated shape electrode when the size becomes smaller, and it takes a lot of time to form the electrode, which makes the process inefficient. On the other hand, in the case of wire EDM, the corner of machined hole becomes round, since the wire electrode has a circular section. Furthermore, in the case of thinner wire electrode, it is not easy to handle. From the above points of view, a new micro EDM technique for machining various fine complicated shape holes with a simple shape electrode is proposed. In this method, a slender electrode with triangular section, which was made by extrusion with fine precision die, is efficiently used.

2. Electrode Forming Method

In micro EDM process, the possibility of fine precision machining significantly depends on the possibility of electrode forming, since the shape of electrode is reversely copied to workpiece through a narrow gap. In general, the electrode is formed by mechanical machining such as cutting, grinding, or reverse discharge method³⁾, in which the electrode is EDMed using block. Recently, wire electrical discharge grinding(WEDG) method⁴⁾ has partially been used. However, these methods are not efficient, since it takes a lot of time to form and only a little part of the electrode is used for EDM. In this study, micro EDM of various fine complicated shape holes by combining the hole machined with a simple shape electrode is proposed.

Fig.1 is the schematic diagram of electrode forming process by extrusion with fine precision die. At first, fine precision die which has triangular section hole is made by micro wire EDM and subsequent finishing process. Next, an electrode material is put into a container and high pressure is applied. Then, a slender electrode with triangular section is extruded through the die. The shape of die can be well copied to the electrode, since silver used as electrode material in this study has fcc structure which leads to high ductility. By this method, triangular section electrode can be formed efficiently. **Fig.2** shows SEM micrographs of the electrode. The length is about 150mm and the cross section is an isosceles right triangle, in which the length of shorter side is about 280 μ m. By using this electrode, it is expected to be easy to machine various complicated section hole, compared with circular or rectangular section electrodes.

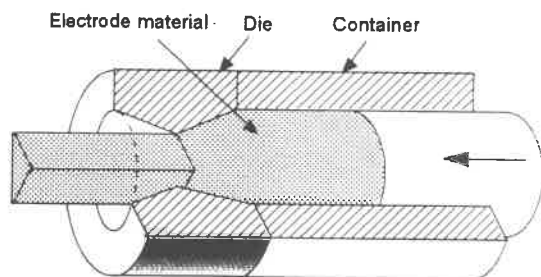


Fig.1 Electrode forming process by extrusion

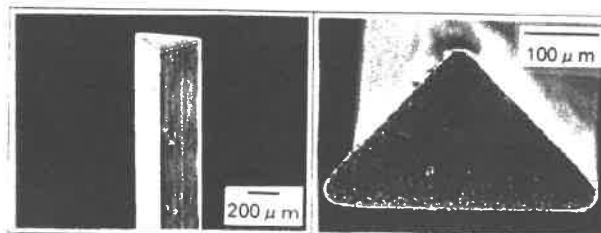


Fig.2 Slender electrode with triangular section

3. Experimental Procedures

Experiments are performed using micro EDM with RC-circuit. Workpiece is set on XY-stage and the machining position can be controlled by a personal computer. Stainless steel SUS304 in JIS specification is used as a workpiece and deionized water (resistivity: $1.0 \times 10^6 \Omega \cdot \text{cm}$) as machining fluid. Machining conditions are set as follows: electrode polarity is negative, open circuit voltage $V=135\text{V}$, condenser capacitance $Ca=2200\text{pF}$.

4. Results and Discussions

4.1 EDM of Fine Complicated Hole

In the first place, machining of straight line was tried. **Fig.3** shows the micrographs of machined straight line. At first, a triangular section hole is machined. Then, the electrode is moved through $100 \mu\text{m}$ parallel to longer side direction of the triangle and a next hole is machined. By repeating this process, straight line of 1.5mm in length was machined. As shown in the photographs, the border between two machined holes is indistinct. It was made clear that straight line could precisely be machined by this method. Then, the machining of complicated shape hole was investigated. **Fig.4** shows examples of regular polygon. The hatched areas as shown in the figure are machined parts, and these examples could be obtained by five or six times machining of triangular section hole. If the section shape of electrode and the central position of the electrode rotation are previously known, various fine complicated holes can be machined effectively and easily.

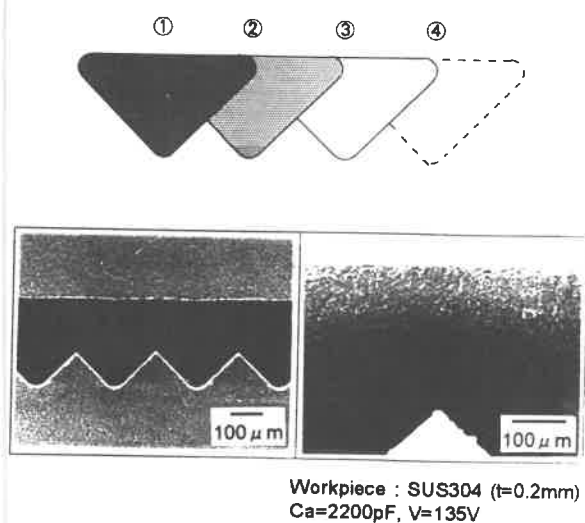


Fig.3 Machining of straight line

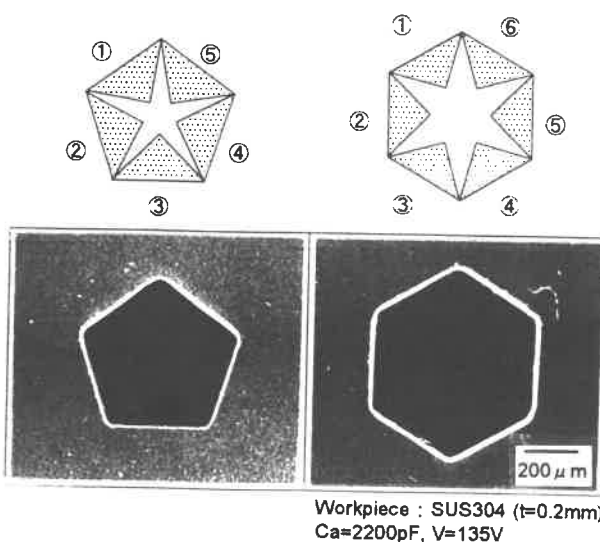


Fig.4 Examples of fine complicated shape hole

4.2 EDM with Reformed Electrode

As mentioned above, various complicated holes can be machined with a simple shape electrode in this technique. However, the roundness more than $25\text{ }\mu\text{m}$ in radius exists at the corner of machined hole, since the electrode formed by extrusion with fine precision die has roundness of about $25\text{ }\mu\text{m}$. Then, reformation of electrode on the machine was tried in order to machine sharper corner hole. Reforming method of the electrode is shown in Fig.5. Copper block is set as shown in the figure, and the electrode is reformed using the reverse discharge method. In this method, the electrode can be reformed easily, since removal amount of anode is much larger than that of cathode in EDM using RC circuit. Fig.6 shows the reformed electrode and the machined hole. The corner of reformed electrode becomes sharper than that of unreformed, and the roundness at the corner of machined hole with the reformed electrode is about $15\text{ }\mu\text{m}$ in radius. Fig.7 shows the machining of star-shape hole using the electrode reforming method. Star-shape hole was generated by five times machining of triangular section hole with a tip angle of 36 degrees. From these results, it was made clear that more complicated shape hole could be machined precisely with the electrode reformed on the machine.

4.3 EDM of Fine Complicated Cavity

Next, machining of fine complicated shape cavity was tried. In the machining of cavity, large undulation of bottom surface of the cavity tends to occur due to electrode wear as shown in Fig.8(a). Therefore, reducing the undulation was attempted by an parallel oscillation between electrode and

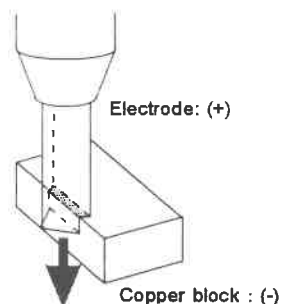


Fig.5 Reforming method of electrode

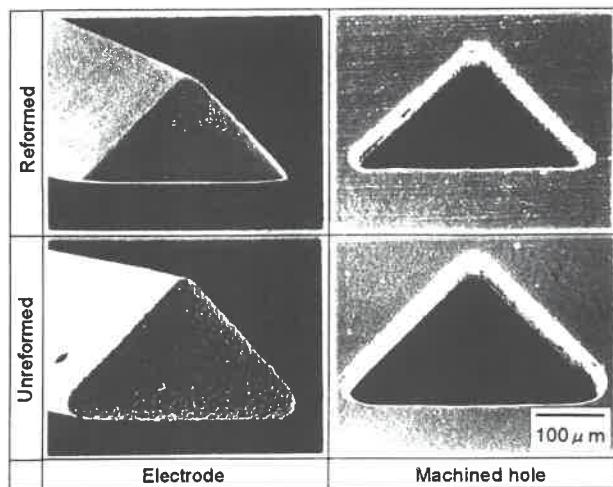
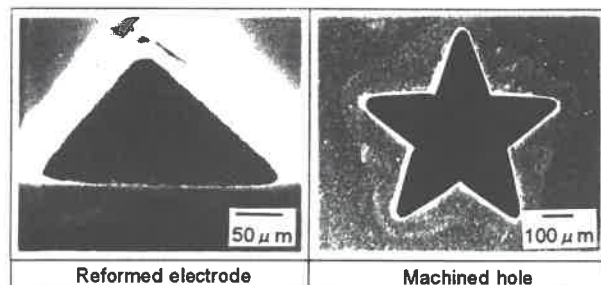


Fig.6 Reformed electrode and machined hole

Workpiece : SUS304 ($t=0.2\text{mm}$)
Ca=2200pF, V=135V



Workpiece : SUS304 ($t=0.2\text{mm}$)
Ca=2200pF, V=135V

Fig.7 Machining of star-shape hole

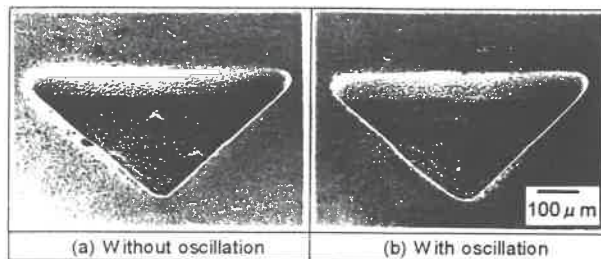
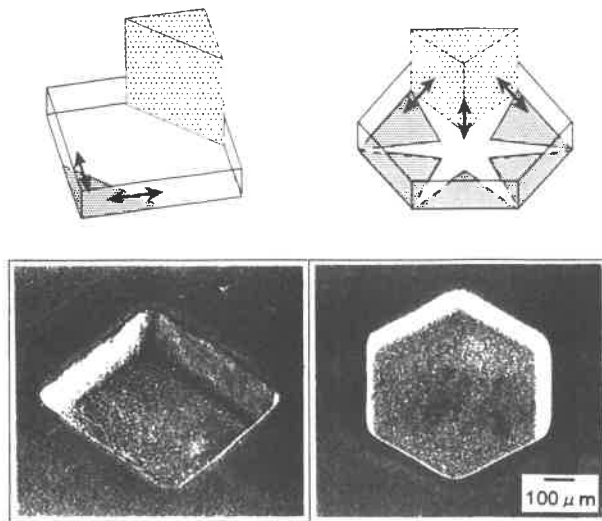


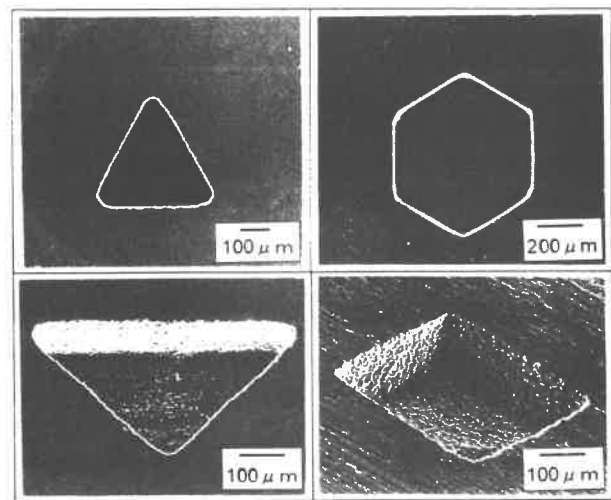
Fig.8 Effect of oscillation

Workpiece : SUS304 ($t=0.2\text{mm}$)
Ca=2200pF, V=135V



Workpiece : SUS304
Ca=2200pF, V=135V

Fig.9 Fine complicated shape cavity



Workpiece : Si
Ca=2200pF, V=135V

Fig.10 Micro machining of monocrystalline silicon

workpiece (stroke:100 μ m, speed:4.9cycle/s). As shown in **Fig.8(b)**, the undulation could be reduced to very small by the oscillation. **Fig.9** shows examples of fine complicated shape cavity. These depths are 150 μ m. The length of one side is 500 μ m in the case of square, and, 380 μ m in the case of hexagon. It was confirmed that fine complicated shape cavity could also be machined by this method.

4.4 Micro EDM of Silicon

Moreover, application of this technique to monocrystalline silicon (resistivity : $10^{-2} \Omega \cdot \text{cm}$) was discussed. Monocrystalline silicon is one of the most important materials in semiconductor industry, and the machining of three-dimensional shape is required. **Fig.10** shows examples of fine complicated shape hole and cavity. Fine holes and cavities can precisely be machined similarly to stainless steel. Therefore, this technique is also effective for micro machining of monocrystalline silicon.

5. Conclusions

Main conclusions obtained in this study are as follows:

- (1) By extrusion with fine precision die, triangular section electrode can be made efficiently.
- (2) Various fine complicated shape holes can be obtained easily by combining machined holes with a triangular section electrode.
- (3) Shaper corner complicated shape hole can be machined precisely with the electrode reformed on the machine using the reverse discharge method.
- (4) This technique is also effective for micro machining of monocrystalline silicon.

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The design and manufacturability of an optic array with micron and micro-radian tolerances for a 30 year system life span*

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Abstract

Performance requirements for an optic array led to a unique precision mount design, which incorporates a full edge support of three large square optics to micron and micro-radian tolerances. The sub-system required performance level versus cost over the mount life cycle led to many design trade-offs and resulted in the monolithic cell that passively mounts the optics. Manufacturability of the mount lands to meet both performance and cost requirements was a major design hurdle. The design philosophy employed is to build all the optic mount requirements in a single cell that does not require adjustments to the individual optics as mounted. This philosophy requires additional effort up-front in the manufacturing set-up but lowers the assembly, alignment, operations and maintenance costs over the 30 year system life.

The National Ignition Facility (NIF) being constructed at Lawrence Livermore National Laboratory is a 1.8 MJ high peak power laser focusing 192 laser beams on a 10 mm target. The final optical component is a frequency converter consisting of two potassium dihydrogen phosphate (KDP) crystals to convert light from 1053 μm to its third harmonic at 351 μm and the focusing lens. The Final Optics Cell (FOC) houses two 41 cm square KDP crystals and a 43 cm square focusing lens with fully supported edge mounts. The square optics are required to minimize space requirements for 192 beamlines.

Issues with the mount design include: maintaining the surface figure of the 2 μm flat 41 cm square crystals, minimizing stresses induced to the crystal (stress optic effects), gravity sag, cleanliness, damage to the crystals and inspection of the crystal as mounted. Prototype results show the manufacturability of the final optics cell design. The first prototypes were fabricated to $4.25 \pm 0.25 \mu\text{m}$ surface figure over the mount lands at commercial vendors using conventional machine tools with no added machine tool environmental controls. This was done with careful characterization of the machine tool and controlling the machining process. The final land surface figure, which has been shown not to cause any distortion or stress into the crystal, will be improved on successive prototypes to a 3-4 micron surface figure. Interferometry and finite element analysis show the stress optic and surface figure variation effects on the frequency conversion process. Performance results of the cell on Beamlet, a demonstration facility for one beamline of NIF, show the performance of the converter design.

Key words: frequency conversion, manufacturing, optic mounts, National Ignition Facility.

1. Introduction

A key design challenge for the National Ignition Facility (NIF), being constructed at Lawrence Livermore National Laboratory (LLNL), [Hibbard, R. L., et. al., 1998], is the frequency converter consisting of two KDP crystals and a focusing lens. Frequency conversion is a critical performance factor for NIF and the optical mount design for this plays a key role in meeting design specifications. The frequency converter, or FOC, Figure 1, is a monolithic cell that mounts the optics and is the point on the beamline where the frequency conversion crystals are optimally aligned and the FOC is focused on target. The lasing medium is neodymium in phosphate glass with a fundamental frequency (1ω) of 1.053 μm . Sum frequency generation in a pair of conversion crystals (KDP/KD*P) produces third harmonic light (3ω or $\lambda=0.35 \mu\text{m}$). The phase-matching scheme on NIF is type I second harmonic generation followed by type II sum-frequency-mixing of the residual fundamental and the second harmonic light. This laser, unlike previous laser system designs, must achieve high conversion efficiency, 85%, which is close to the 90.8% theoretical maximum and thus drives the optic mount design. As a

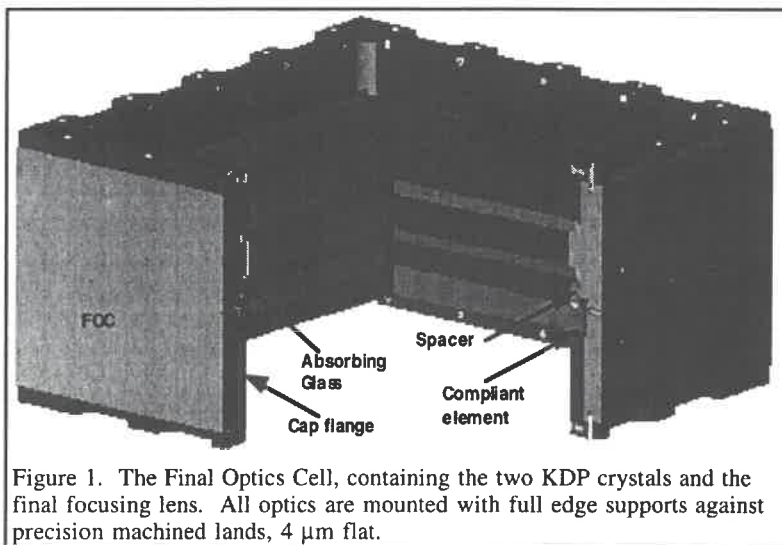


Figure 1. The Final Optics Cell, containing the two KDP crystals and the final focusing lens. All optics are mounted with full edge supports against precision machined lands, 4 μm flat.

result, this design is very sensitive to angular variations in beam propagation and in the crystal axes orientation. These angular variations need to be controlled within a 40 μ rad error budget. The optical mount contributions to the angular error budget are what make the FOC such a challenging precision design.

The full edge support used in the FOC design is driven by the spherical target chamber that has optics mounted at multiple longitudinal angles spanning 170° and thus gravity sag in the crystals that needs to be minimized. To meet the angular performance requirements, a precision monolithic cell with full edge support for mounting the optics to 10 μ rad angular and 4 μ m flatness tolerances is required. The NIF frequency converter design is a major step in improving both conversion efficiency and precision of the optical mount.

2. Design Requirements

The design requirements for the NIF final optics fall into two categories: the overall system level requirements; and the driving requirement to optimize the frequency conversion efficiency. Both requirements will be discussed, however the main focus of this paper will be on the mount design specifics affecting conversion efficiency. Figure 2 shows the Final Optics Assembly (FOA), which houses the FOC and its actuation system. The upper portion of the FOA, consists of four integrated optics modules that provide a clean barrier from the target chamber to the Class 100 final optics environment. Table 1 shows the stringent system alignment and motion tolerances, and cleanliness requirements to optimize the optics performance.

The angular error budget, 40 μ rad internal angle, for frequency conversion efficiency includes index inhomogeneity of the crystals, alignment, mount induced stresses, gravity sag, surface figure distortions, and metrology, [Wegner, P. J., et. al., 1998]. 20 μ rad is allocated to the combined mounting affects. Other design requirements are derived requirements such as a 3 point kinematic support to tie the FOC to the actuation system to minimize the stresses induced to the cell to under 50 psi and distortions to the mounting lands of less than 0.5 μ m.

Table 1. System requirements for the FOC

Requirement	Specification	Reason
Crystal and lens mounting surface fig.	3-5 μ m	Frequency conversion (FC) efficiency
Angle between crystals and the lens	± 10 and ± 12 μ rad	FC and alignment
Dynamic stability for final alignment	± 3.1 μ m lateral and ± 4.6 μ m angular	Alignment
Cleanliness	Class 100	Optics damage
Vacuum	1.0e-5 torr	Experiment req.
Temperature	± 0.1 C°	FC sensitivity
Angular resolution/acc.	± 2 μ rad and ± 5 μ rad	Alignment and FC
Linear resolution/acc.	± 0.1 mm & ± 0.3 mm	For focusing on target

3. The FOC Design

The FOC design philosophy is to passively build in all the required mount performance requirements. This ensures long-term efficiencies in assembly, alignment, assembly, operations and maintenance. Then a cell only needs to be assembled correctly and checked; there is no need or possibility to adjust the figure of the three individual optics components inside the cell. This also allows a much simpler design which lowers the cell cost. If one considers trying to individually adjust the optics positions, surface figure and angular tolerances on 192 beam lines during operations this passive cell design becomes very attractive. The philosophy is if all requirements are built into the cell the problem reduces to a manufacturing issue for the cell. The target cost for manufacturing the cells (quantity 192) was set at \$7,000 per cell. Prototype results have validated this number.

Figure 1 shows that the crystals are mounted against a precision machined surface, and on the other side is a compliant element. The load is set by the shim size between the cap flange and the top of the cell; the spacing is individually calculated for each set of mating parts. The absorbing glass around all the edges of the crystals is NG4 glass to absorb the Stimulated Raman Scattered light from the crystals. The cell is kinematically mounted on a 120° spacing around the cell. This mount design minimizes the stresses and displacements transferred to the cell, in the worst case the distortions to the cell are in the sub micron range. Finite element analysis shows the cell natural frequencies are above 100 Hz.

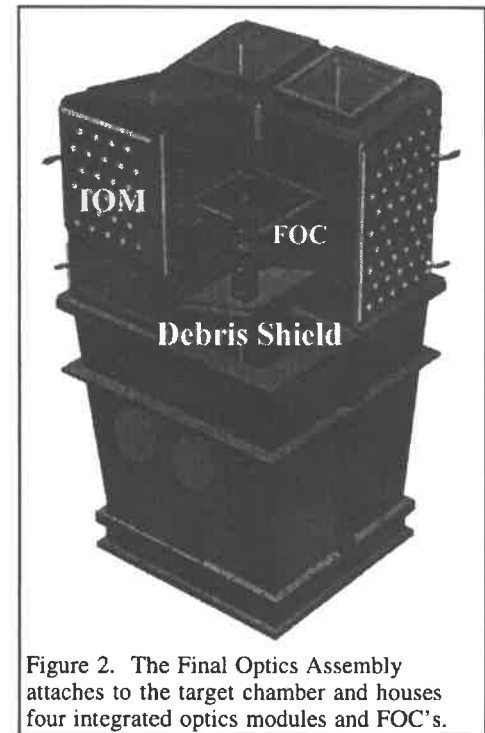


Figure 2. The Final Optics Assembly attaches to the target chamber and houses four integrated optics modules and FOC's.

If there was only one beamline, there are many ways one could achieve a 1-5 μm flat mount, but since there are 192 beamlines cost is an issue. Thus, for this design the manufacturability of the FOC became a design driver. For a single cell one could consider manufacturing methods such as replication, diamond turning or lapping mount surfaces and bonding or bolting the pieces together, but these were rejected for cost reasons. If the mounting surface only had to be 12 μm flat the manufacturing of the cell could be done by many high quality machine shops. Conversely if the cell had to be 1 μm flat an ultra-precision method of producing the cell would be needed and the cost of producing the cells would be very high. The interesting question is in-between 1 and 12 μm what can be done at a reasonable cost on a conventional machine tool?

Another factor to consider in the design is the material properties of the KDP crystals, a low modulus and yet brittle material, which makes the handling, and cleanliness of them a concern for the mount design. Because of the small area of the mount, 5 mm by 41 cm on four sides, and the loading of the crystal against a hard surface, cleanliness becomes a critical concern. A five micron particle on the mount could cause both a distortion in the surface figure locally and put the mounted part out of specification and also could be the cause of a chip in the crystal due to localized point loading. These minute flaws can also cause induced stresses in the crystals which cause a drop in conversion efficiency, because of a loss in the index of refraction within the crystal.

4. Prototyping Results

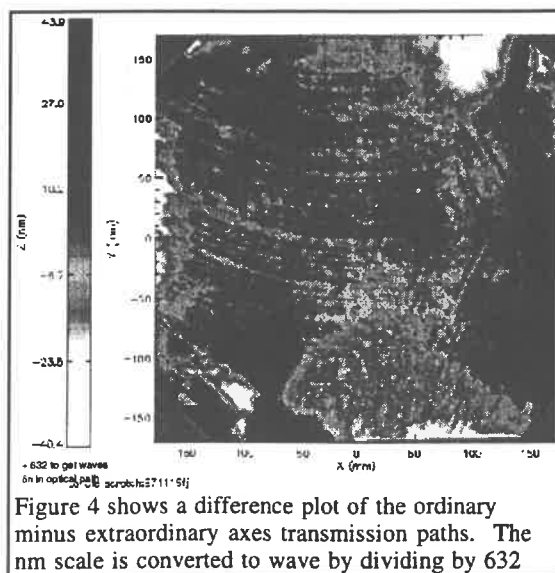
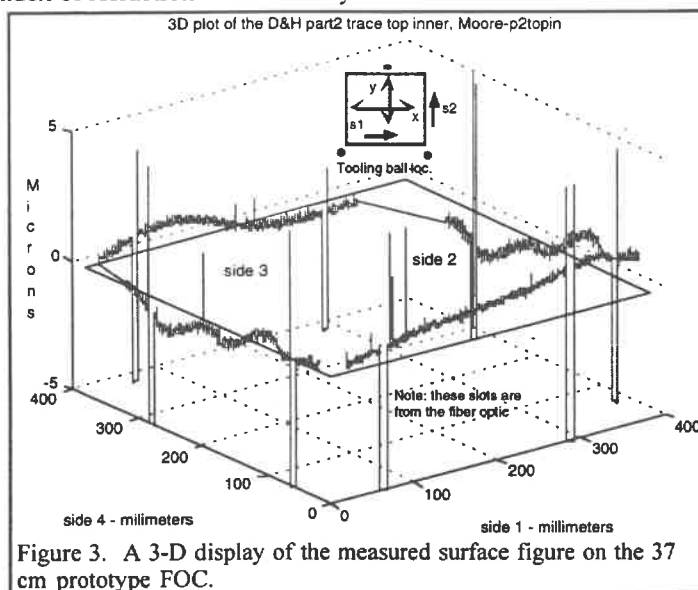
Prototype results discussed here will include the manufacturing of the FOC's, off-line metrology to qualify as mounted crystals and performance results from Beamlet a prototype beam of the NIF laser.

Manufacturing - The FOC development effort was focused on conventional CNC machines for cost reasons. The mandate for producing the first FOC's was on a best effort basis, under 10 μm flat and parallel with the goal of 3-7 μm flatness. The plan was to do the best we could with the standard commercial equipment and with a LLNL precision machinist helping in characterizing the machine and providing input into the manufacturing methodology.

Figure 3 shows a 3-D plot of a measurement of the top flange made at LLNL on a Moore M5 measuring machine with an accuracy of $\pm 0.25 \mu\text{m}$, using an air bearing LVDT. This figure shows the part measures flat within 4.25 μm which is an outstanding result considering the environment the machine tool was in. The significant result is the machined surface is very close to the machine geometry which is the best that could be expected. A significant development hurdle has been crossed; FOC's can be machined to the required tolerances of 3-4 μm cost effectively at a commercial vendor.

Crystal as mounted performance results - the key measurements are surface figure deformations and induced stress effects. Presently, interferometry, both transmitted wave front (TWF) and reflected wave front, (RWF) measurements, are being used. The TWF measures transmission losses in the crystal and when combined with frequency conversion modeling allows one to predict the induced stress effect on conversion efficiency. RWF gives a measure of the crystal surface figure deformations and edge flaws in mounting the optic.

Figure 4 shows a plot of the difference between TWF measurements in the ordinary and extraordinary axes on a 37 cm doubler in the FOC. This plot shows crystal bulk property losses in transmission and gives a measure of the changes in $\delta n = \delta n_o - \delta n_e$, the index of refraction, assuming constant crystal thickness. Looking at the bottom



left corner shows a region with a bow-tie like feature, to the right is a large triangular region, and in the upper right corner a crescent shaped feature. All of these are due to internal growth imperfections within the crystal. The change in the index of refraction is related to stress effects, both induced and from growth.

Figures 5 (a and b) show RWF measurements made on the same crystal in the FOC with a 1.0 lb/in line load and with the cap flange loosened all the way, which approximates a free-standing measure.

Figure 5 (a) shows the free-standing wavefront is $2.75 \mu\text{m}$ and has a general saddle shape.

Figure 5 (b) shows the loaded crystal surface figure is $3.2 \mu\text{m}$, but

the dominant shape has changed from a saddle to a more cylindrical shape where the bottom left and upper right corners have bent up. This implies the mounting is not altering the surface figure on a gross scale, but the changes in surface figure do cause small local phase matching angle errors.

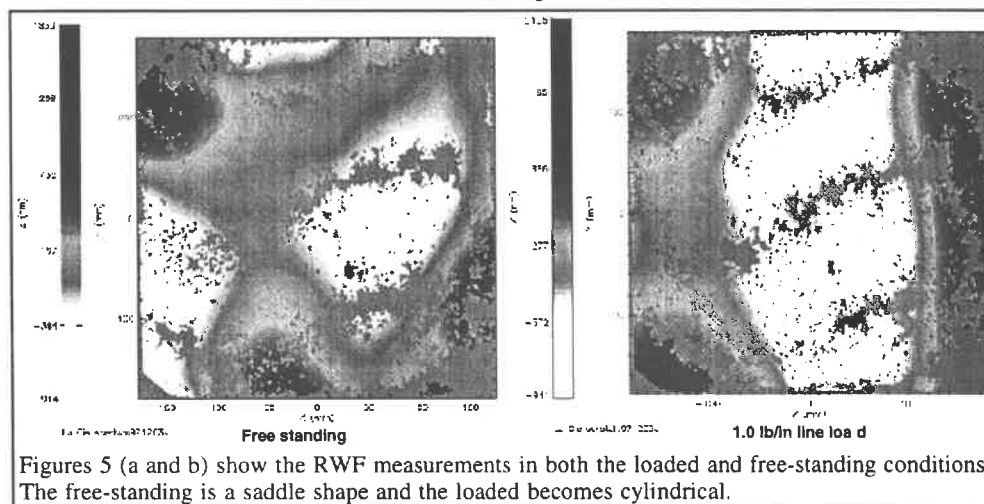
A non-linear finite element analysis (FEA) model has been built to model and predict the details of the contact interface between the mount surface, the crystal and the compliant element. Results show that at line loads of 0-1 lb/in the crystal is contacting high points on the mount but not fully taking the shape of the mount, Figure 6. As the load is increased to 2.0 lb/in the crystal fully takes the shape of the cell and distorts the surface figure excessively. This correlates with interferometry. As a result the design loads have been decreased to 0.6 – 1.0 lb/in.

The FOC has been installed on Beamlet and frequency conversion measurements have been done to verify results from both off-line metrology and frequency conversion modeling. From on-line interferometry measurements the detuning angle on a 32 cm beam was calculated from the slope of the surface figure in the sensitive vertical direction and dividing by the refractive index yields the map of the detuning caused by surface refraction. The double detuning inferred from the surface figure has only 5% of the beam area outside of a $\pm 22 \mu\text{rad}$ angle. This shows the mounted performance of the crystal in the cell almost meets the 20 μradian error budget on this prototype setup.

5. Summary

The FOC mount design is critical to the optical performance of the frequency conversion crystals. Results show influences of the line loads, surface figure and induced stresses on frequency conversion do not degrade the optical performance. The monolithic FOC cell design has many advantages in its simplicity and how it passively builds in all the desired accuracy for mounting the optics. Manufacturability has been proven at a level of $4 \mu\text{m}$ of surface figure. Further work is needed to finalized the specifications on the required flatness and performance of the mount.

*** This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract No. W-7405-Eng-48.**



Figures 5 (a and b) show the RWF measurements in both the loaded and free-standing conditions. The free-standing is a saddle shape and the loaded becomes cylindrical.

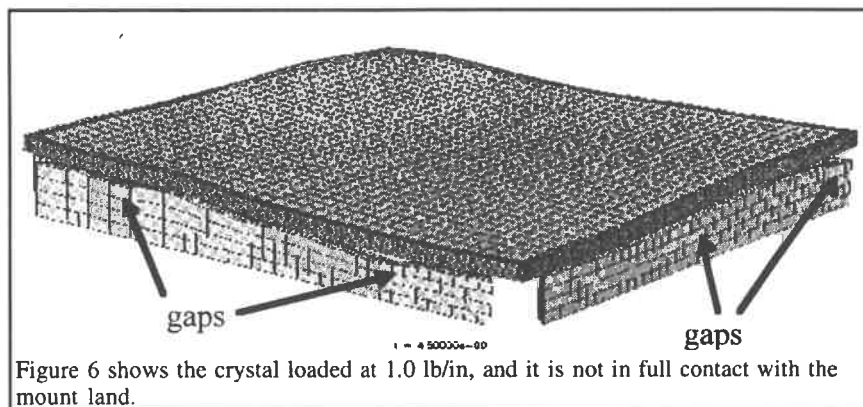


Figure 6 shows the crystal loaded at 1.0 lb/in, and it is not in full contact with the mount land.

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FABRICATION OF A HIGH SPEED MOBILE MICROMECHANISM FOR CURVED SMALL-DIAMETER PIPES

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Abstract

A new high speed micromechanism, which is 6.2 mm wide, 88 mm long, 6.8 g in the mass and imitating the motion of an inchworm driven by pneumatic pressure, is fabricated. The micromechanism is confirmed to move in the thin curved-pipes of 12 mm in diameter and 150 mm in curvature with the speed of 46 mm/s.

Keywords: High speed, In-pipe, Mobile, Micromechanism, Inchworm-motion

1. Introduction

New micromechanisms to inspect or to repair thin pipes in the the human body or pipelines have been searched. Takahashi et al. have presented a flexible inching mechanism which is 20 mm in diameter and able to inch in pipes by imitating the creeping motion of an earthworm [1]. Yoshida et al. also have presented a new micromechanism which is made of a rubber tube and a spring of 3 mm in diameter and driven by water pressure [2].

However, moving speeds of these micromechanisms are not sufficient for the micromechanisms to inspect or to repair thin pipes in the the human body or pipelines, because the speeds of the micromechanisms are 2 or 7 mm/s.

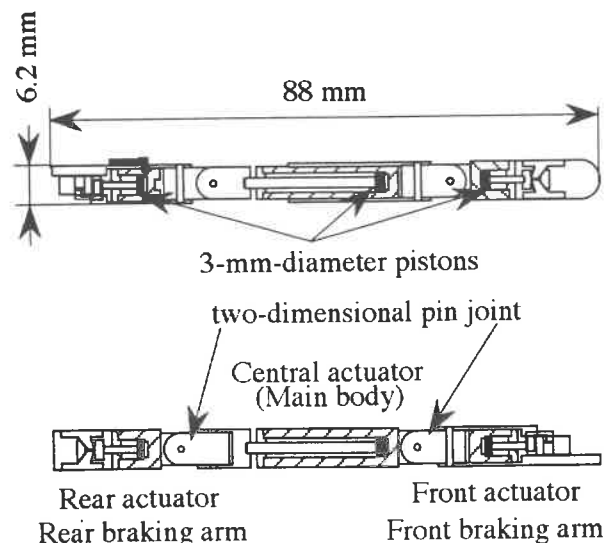


Fig. 1 The fabricated micromechanism (section)

We have fabricated a new high speed micromechanism which is driven by pneumatic pressure. The micromechanism is able to move in pipes by imitating the motion of an inchworm. It is able to move in the thin curved-pipes of 12 mm in diameter and 150 mm in curvature with the speed of 46 mm/s which is more than six times of that of former in-pipe mobile micromechanisms.

2. Structure of the fabricated micromechanism

The section of the fabricated micromechanism is shown in Fig. 1. The action of the parts of the micromechanism is shown in Fig. 2. The micromechanism is able to move in pipes by imitating the motion of an inchworm. The micromechanism is constructed by a front braking arm, a front actuator for the front braking arm, a central actuator (main body) for the inching motion of the micromechanism, a rear braking arm, a rear actuator for the rear braking arm and two two-dimensional pin joints. The parts of the micromechanism are made of aluminum. Each actuator has a piston of 3 mm in diameter and a return spring. The return spring for the braking arm is 2.6 mm in free length and has the spring stiffness of 0.147 N/mm.

The return spring for the central actuator (main body) is 18.3 mm in free length and has the spring stiffness of 0.098 N/mm.

The micromechanism is able to move in the curved-pipe, because it has two two-dimensional pin joints at the both side of the central actuator. The micromechanism is 6.2 mm wide, 88 mm long and 6.8 g in the mass. Three plastic air-tubes which feed pneumatic pressure to the micromechanism are 0.5 mm in inner diameter, 1 mm in outer diameter, and 400 mm long.

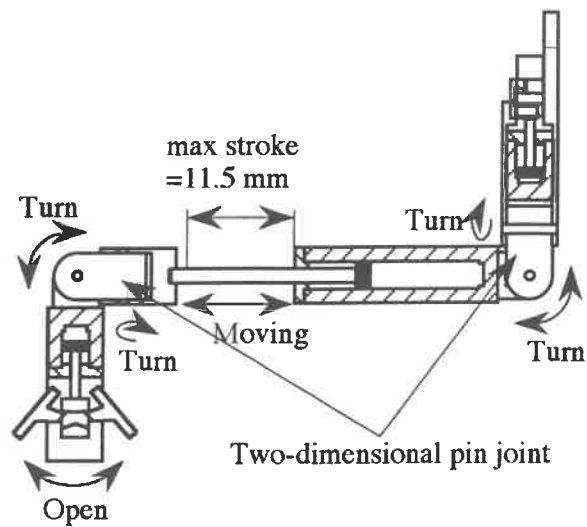


Fig. 2 The fabricated micromechanism (action of the parts)

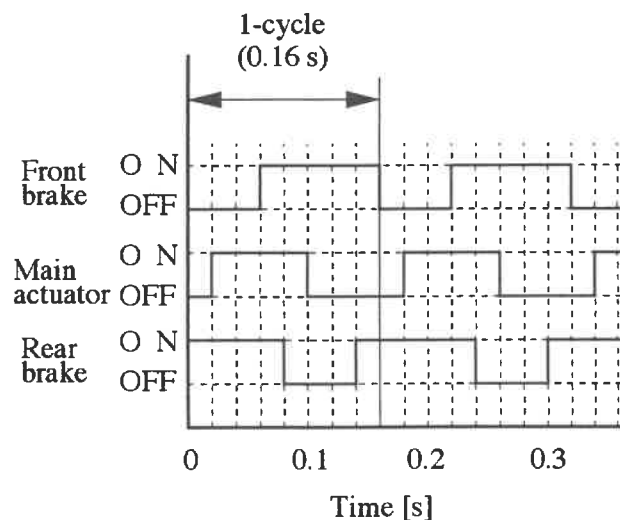


Fig. 3 The time-chart of the pneumatic pressure

First, the rear braking arm is opened by the rear actuator and the micromechanism is held in the pipe. Next, the central actuator is acted and the micromechanism is moved by stretching motion of the central actuator. Then, the front braking arm is acted by the front actuator and the micromechanism is held in the pipe. At last, the rear braking arm is closed to be free and the central actuator is shrunk. Consequently, the micromechanism is achieved to inch. We used the cycle time of 0.16 seconds that is needed to act three pneumatic actuators to get the maximum inching speed. The time-chart of the pneumatic pressure for the micromechanism is shown in Fig. 3.

3. Moving characteristics of the fabricated micromechanism

The experimental apparatus for the measuring of the moving speed is shown in Fig. 4. The pneumatic pressure of 0.3 MPa is fed for the braking actuators. For the central actuator, pneumatic pressures from 0.3 MPa to 0.5 MPa are fed. A thin straight pipe of 12 mm in diameter and a curved-pipe of 150 mm in curvature are used for the small-diameter pipes. The sloping angles of the pipes are changed from 0° to 50°.

The relationship between moving speed of the micromechanism in a thin straight pipe of 12 mm in diameter and the sloping angle of the pipe is shown in Fig. 5. The moving speed of the micromechanism is lowest at the pressure of 0.3 MPa. However, there is a little difference of speed between the pressure of 0.4 MPa and 0.5 MPa. The maximum speed at the horizontal angle has been obtained as 46 mm/s at the pressure of 0.4 MPa.

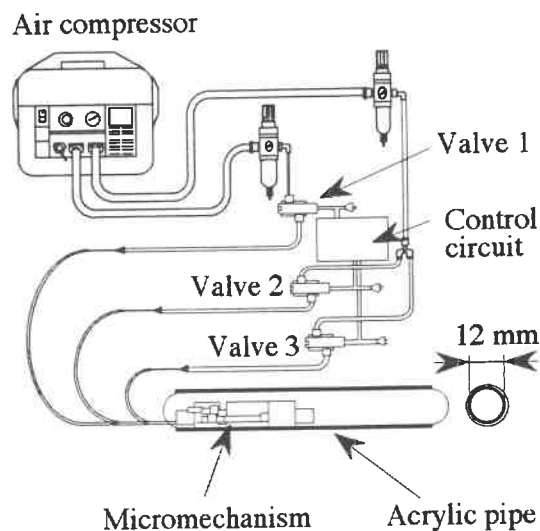


Fig. 4 Experimental apparatus

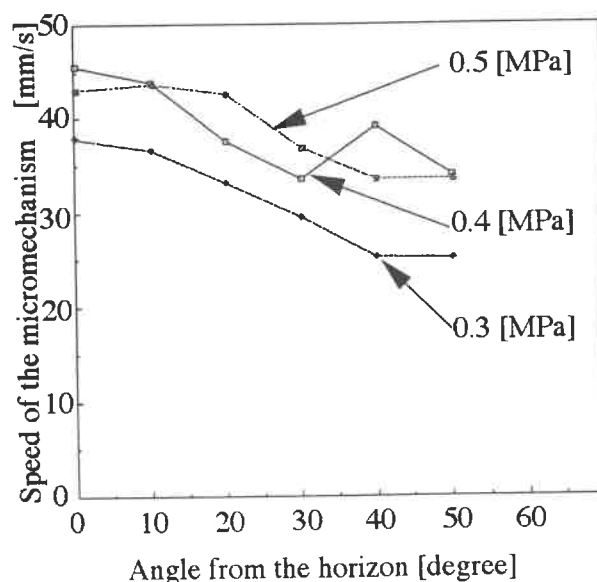


Fig. 5 Speed of the micromechanism (Straight pipe)

The relationship between moving speed of the micromechanism in a thin curved-pipe of 150 mm in curvature and the sloping angle of the pipe is shown in Fig. 6. The maximum speed at the horizontal angle has been also obtained as 46 mm/s at the pressure of 0.4 MPa. These speeds are more than six times of that of former in-pipe mobile micromechanisms.

It is expected that higher moving speed will be obtained by using more thick feeding air-tubes and a shorter cycle time, because the flow-resistance of air-tubes is inversely proportional to the second or 2.5th power of the diameter of feeding air-tubes.

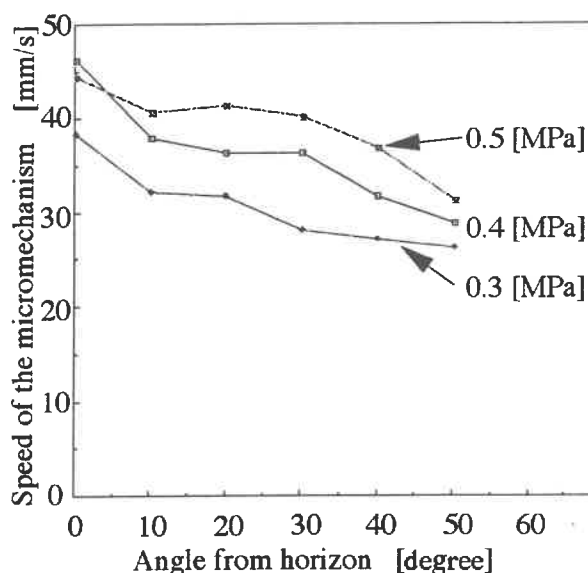


Fig. 6 Speed of the micromechanism (Curved pipe)

4. Conclusions

A fabrication of a new micromechanism to inspect or to repair thin pipes in the the human body or pipelines has been performed. The followings are the conclusions of the fabrication.

1) We have fabricated a new high speed micromechanism which is imitating the motion of an inchworm and driven by pneumatic pressure. The micromechanism is constructed by two actuators for the braking arms, a central actuator for the inching motion of the micromechanism, and two two-dimensional pin joints. The micromechanism is 6.2 mm wide, 88 mm long and 6.8 g in the mass.

2) The micromechanism is confirmed to move in the thin curved-pipes of 12 mm in diameter and 150 mm in curvature with the speed of 46 mm/s which is more than six times of that of former in-pipe mobile micromechanisms.

References:

- [1] M.Takahashi, I. Hayashi, N. Iwatsuki, K. Suzumori and N. Ohki, *Journal of the Japan Society for Precision Engineering*, Vol. 61 No. 1 (1995), pp. 90-94.
- [2] K. Yoshida, K. Takahashi and S. Yokota, *Proceedings of IMPT-100, The Japan Society of Mechanical Engineers* (1997), pp. 667-670.

The Fabrication and Testing of Optics for EUV Projection Lithography^{1,2}

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Introduction

Extreme UltraViolet Lithography (EUVL) is a leading candidate as a stepper technology for fabricating the next generation of microelectronic circuits. EUVL is an optical printing technique qualitatively similar to Deep UV Lithography (DUVL), except that 11-13nm wavelength light is used instead of 193-248nm. The feasibility of creating 0.1 μ m features has been well-established using small-field EUVL printing tools, and development efforts are currently underway to demonstrate that cost-effective production equipment can be engineered to perform full-width ring-field imaging consistent with high wafer throughput rates. Ensuring that an industrial supplier base will be available for key components and subsystems is crucial to the success of EUVL. In particular, the projection optics are the heart of the EUVL imaging system, yet they have figure and finish specifications that are beyond the state-of-the-art in optics manufacturing. Thus it is important to demonstrate that industry will be able to fabricate and certify these optics commensurate with EUVL requirements. The goal of this paper is to demonstrate that procuring EUVL projection optical substrates is feasible.

Specifications

EUVL projection systems employ all-reflective configurations in a ring field geometry with the mirror surfaces having stringent specifications on both figure and finish.³ Nominal specifications for the average allowable figure and finish errors on individual substrates for a 4-mirror EUVL projection optics design are given in Table 1. These multi-mirror systems typically utilize aspheric surfaces to obtain aberration correction, which adds a significant degree of difficulty to the fabrication and testing of the substrates.

The three specification categories listed in Table 1 are tied closely to performance requirements. The figure specification is motivated by the requirement for a high resolution imaging system with low distortion across the image field. If the projection optics are to achieve diffraction-limited performance by Marechal's criterion, the composite wavefront error (at the exit pupil) must be less than $\lambda/14$ rms where λ is the operating wavelength ($\lambda = 13.4$ nm). In the context of our 4-mirror system, each surface should contribute, on the average, no more than 0.5 nm rms of WFE (assuming the errors are uncorrelated), which leads to the average figure error of 0.25 nm rms specified in Table 1. Mid-spatial frequency roughness (MSFR) (spatial periods of 1 μ m to 1 mm) causes near-angle scattering where the scattered light remains in the image field. Thus, MSFR causes a background illumination, usually

Table 1. Nominal specifications for EUVL projection optics (4 mirror system).

Error Term	Maximum Error Specification	Defined by integrating the Power Spectral Density (PSD) of surface errors over the following bandlimits:
Figure	0.25 nm rms	(Clear Aperture) ⁻¹ - 1 mm ⁻¹
Mid-Spatial Frequency Roughness (MSFR)	0.20 nm rms	1 mm ⁻¹ - 1 μ m ⁻¹
High-Spatial Frequency Roughness (HSFR)	0.10 nm rms	1 μ m ⁻¹ - 50 μ m ⁻¹

referred to as *flare*, that is superimposed on the desired image. The most prominent effect of flare is to reduce image contrast, which adversely limits the range of acceptable operating conditions (process window) for performing lithography. In addition, if the flare is non-uniform over the image field, then the critical dimensions (CD) of printed features will also exhibit non-uniformity. Modeling efforts indicate that acceptable levels of image contrast and CD variation can be achieved for MSFR levels of 0.2-nm rms, as specified in Table 1. Wide-angle scattering is caused by high-spatial frequency roughness (HSFR) (spatial periods $<1\ \mu\text{m}$) and results in a loss to the system because light is scattered outside of the image field. Wide-angle scattering also decreases contrast by reducing the intensity of the "light" areas of the image field, in comparison with near-angle scattering, which decreases contrast by redistributing energy within the image field.

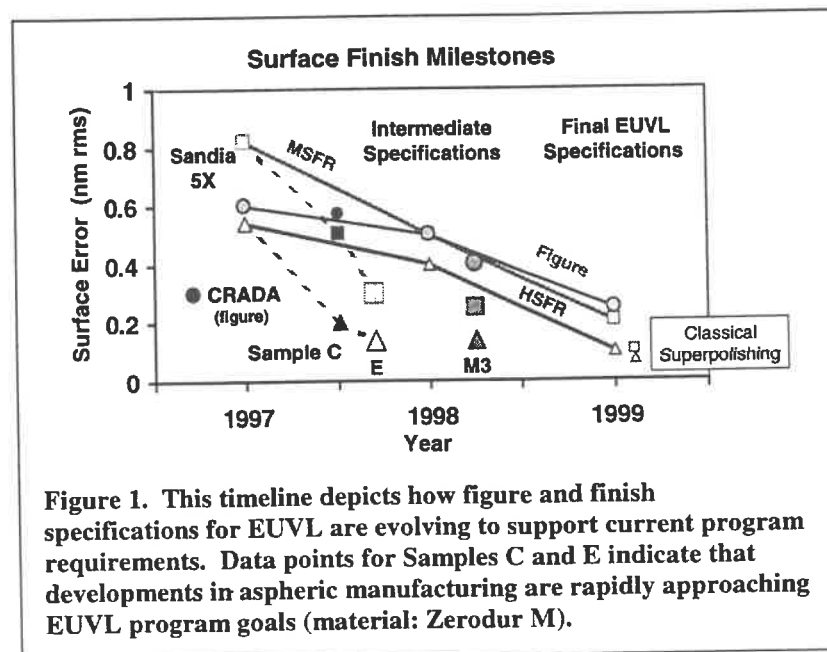
The Evolution of Optical Fabrication to Meet EUVL Specifications

Figure 1 is a plot of surface errors for the three key specification categories (figure, MSFR, HSFR) versus the year in which they should be achieved in order to meet current program requirements; each of the three lines corresponds to a different category. The three points at 1997 ("Sandia 5x") are the measured values from aspheric surfaces fabricated for another project and represent the state-of-the-art for combined figure and finish at the beginning of our current program. The fact that the MSFR is greater than the figure error is indicative of the impressive level of determinism in the figuring process, while also indicating that finish is often sacrificed during aspheric processing. In comparison, the data point labeled "CRADA" for mid-1996 represents an impressive 0.3 nm rms measured figure error, although finish was not specifically controlled for that optic.⁴ The points indicated for 1998 and 1999 are the specifications for optics required in the current effort; the *Intermediate Specifications* in 1998 offer an interim milestone mid-way during the finishing process development that links the state-of-the-art in 1997 with the *Final EUVL Specifications* for 1999. Complete sets of optical elements will be delivered and evaluated for both the intermediate and final specifications.

The two points to the right of 1999 labeled as "Classical Superpolishing" are the levels of MSFR and HSFR that can be produced on flats and spheres by companies that perform "superpolishing", such as for laser gyro components. These points represent a "proof of existence" that demonstrates that optical fabrication methods can indeed attain these levels of finish.

There is a growing effort in the optics community to improve aspheric fabrication technology in order to meet the future needs of EUVL. As part of our efforts in identifying and reducing program risks, we are assessing this rate of improvement by purchasing optical substrates from commercial supplier(s) using specifications that become increasingly stringent, as shown in Figure 1.

Our test and evaluation samples have been obtained from commercial source(s) and have all been fabricated using aspheric processing methods, although the surface contours are flat or spherical in order to simplify inspection. Some of the substrates have had simultaneous figure and finish specifications, while others have concentrated on improving finish while using aspheric methods. In all cases, the required specifications have been met.



The results for two of the developmental samples are shown in Figure 1. Sample C is a spherical substrate fabricated using aspheric methods with specifications for figure, MSFR, and HSFR. The key observation is that even while maintaining figure quality to 0.6-nm rms, the values of MSFR and HSFR dropped to 0.51 nm rms and 0.20 nm rms, respectively. The goal of flat Sample E was to focus aspheric processing development on improving surface finish, without a stringent specification on figure. The MSFR value of 0.30-nm rms and HSFR value of 0.14 nm rms are plotted on the timeline in Figure 1. Both of these values are closely approaching the final EUVL specifications.

An important milestone was achieved by the fabrication of the initial element of our 4-mirror projection system to the *Intermediate Specifications* shown in Figure 1. Mirror M3 is a spherical Zerodur M substrate that was fabricated using the aspheric methods developed with the developmental samples. Aspheric finishing techniques were used instead of traditional spherical polishing methods because of their demonstrated determinism in meeting stringent figure requirements. For comparison, the figure and finish tolerances that were achieved are indicated in Figure 1, by the three symbols to the right of the *Intermediate Specifications*. A figure error of 0.4 nm rms, MSFR of 0.22 nm rms, and a HSFR of 0.14 nm rms were achieved, which are significantly below the specified values. Completion of this element demonstrates the feasibility of meeting stringent specifications using aspheric methods. As this manuscript is being written, the other elements of this set are being completed.

Calculation of Surface Errors

As noted in the definition of the specifications given in Table 1, the relevant parameter is the rms power obtained by integrating the 2-D power spectral density (PSD) over an appropriate band of spatial frequencies. The use of 1-D PSDs is well described in the literature, particularly in relating light scattering to surface statistics.⁵ However, the calculation of statistics from the 2-D PSD is relatively uncommon, although there is a trend for its increasing application. In order to calculate the relevant surface statistics, we calculate the 2-D PSD, and then delineate a circular region within the frequency domain with a radius of the Nyquist frequency (or an ellipse if the Nyquist frequencies are different in the x and y directions).⁶ We then integrate the 2-D PSD in the annular region that includes the spatial frequencies that are relevant. For example, we would integrate between the frequencies of 1/mm to 1/ μ m in order to assess the MSFR. Measurements may be required from more than one instrument in order to span a sufficiently wide frequency region. PSDs calculated from surface roughness measurements have shown excellent agreement with PSDs inferred from angle-resolved scattering data.⁷

Metrology for EUVL Optics

The bandwidths of the three specification categories given in Table 1, figure, MSFR, and HSFR, correspond approximately to the bandwidths measured by three types of instruments: full-aperture phase-shifting interferometry, phase-shifting interferometric microscopy (AFM), respectively. Thus, power spectral densities can be assembled that cover an extremely wide spatial frequency band, i.e. CA^{-1} to 0.1 nm^{-1} . Figure 2 shows the average radial PSD for M3 that shows the contributions over about six orders of magnitude of spatial

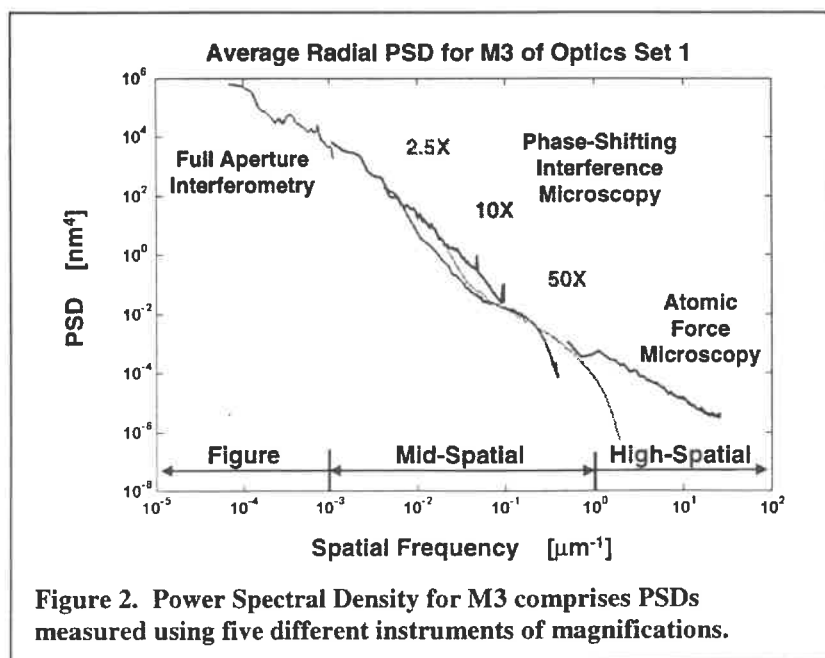


Figure 2. Power Spectral Density for M3 comprises PSDs measured using five different instruments of magnifications.

frequency. Three magnifications were used with the phase-shifting interferometric microscope for measuring the MSFR. It is outside of the scope of this summary to provide a comprehensive discussion of these types of measurements. However, the measurement of figure accuracy is an especially important issue, because of the need to measure absolute accuracy with respect to the optical design.

We have designed and built a phase-shifting diffraction interferometer (PSDI) for measuring the figure accuracy of EUVL mirror substrates.⁸ Errors in the reference have been minimized by using two arbitrarily perfect spherical wavefronts: one serves as the measurement wavefront and is incident on the mirror surface under test and the other serves as the reference wavefront. The calculated deviation from sphericity for reference waves generated with a diffracting aperture are of order 10^{-6} waves (visible). Control of their relative amplitude and phase provides contrast adjustment and phase shifting capability. The PSDI has been successfully applied to measuring aspheric surfaces with low fringe density or to more general aspheric surfaces by stitching together regions that are independently measured with low fringe density. This concept has been implemented in several ways using lithographically generated apertures or single mode optical fibers.

Conclusion

The optics industry is currently attaining figure and finish values near those required for EUVL aspheres. The rate at which finishing performance is improving will enable a projection optics system to be assembled for meeting current EUVL program requirements.

Notes and References

¹ **Auspices:** This work was performed under the auspices of the U.S. Department of Energy by the Lawrence Livermore National Laboratory under Contract No. W-7405-ENG-48. Funding was provided by the Extreme Ultraviolet Limited Liability Corporation under a Cooperative Research and Development Agreement.

² For additional references and an expanded description of many topics mentioned in this summary, see: Taylor, J. S., Sommargren, G. E., Sweeney, D. W., and Hudyma, R. M., "The fabrication and testing of optics for EUV lithography", in *Emerging Lithographic Technologies II*, Proc. SPIE vol. 3331, pp. 580-590 (1998).

³ Sweeney, D. W., Hudyma, R. M., Chapman, H. N., and Shafer, D. R., "EUV optical design for a 100-nm CD imaging system", in *Emerging Lithographic Technologies II*, Proc. SPIE vol. 3331, pp. 2-10 (1998).

⁴ Kestner, R., "Precision asphere fabrication and metrology to tolerances <1 nm rms", paper presented at the OSA Optical Fabrication and Testing Topical Meeting, Boston, May 1-3, 1996; note that this measured value represents a comparison with respect to the vendor's interferometric reference, not a definitive statement of accuracy.

⁵ Church, E. L. and Takacs, P. Z., "Specification of glancing- and normal-incidence x-ray mirrors", *Optical Engineering*, vol. 34, no. 2, pp. 353-360 (Feb. 1995).

⁶ The 2-D PSD is the squared magnitude of the 2-D Fourier transform of the surface errors and is calculated as:

$$S(f_x, f_y) = \frac{L_x L_y}{M^2 N^2} \left| \sum_{m=0}^{M-1} \sum_{n=0}^{N-1} z(x_m, y_n) e^{-2\pi i m \Delta x f_x} e^{-2\pi i n \Delta y f_y} \right|^2$$

where $z(x,y)$ is the distribution of surface errors over the x,y plane

L_x, L_y are scan lengths in the x and y directions

f_x, f_y are spatial frequencies in the x and y directions

M, N are the number of equi-spaced samples in the x and y directions

$\Delta x, \Delta y$ are the sample spacings for x and y

⁷ Gullikson, E. M., "Nonspecular scattering for normal-incidence EUV optics", in *Emerging Lithographic Technologies II*, Proc. SPIE vol. 3331 (1998) to be published.

⁸ Sommargren, G. E., "Phase shifting diffraction interferometry for measuring extreme ultraviolet optics", in *Extreme Ultraviolet Lithography, Trends in Optics and Photonics (TOPS)*, vol. IV, G. D. Kubiak and D. R. Kania eds., Optical Society of America, pp. 108-112 (1996).

Design Overview of a Prototype Large Aperture Deformable Mirror for the National Ignition Facility

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Abstract

A large aperture prototype deformable mirror was designed, fabricated and tested at Lawrence Livermore National Laboratory (LLNL) for future application to the National Ignition Facility (NIF). To our knowledge, this development produced the largest aperture continuous surface deformable mirror adaptive optic system ever for a mega-Joule class laser and target irradiation facility. Experimental results, collected while operating the deformable mirror in a laboratory environment and on Beamlet, a NIF single beam prototype, demonstrate the value of a large aperture deformable mirror and that it can be predictably designed, built and operated as an alternative to making a "perfect" optical system.

1. Introduction

The National Ignition Facility (NIF), currently under construction by the Department of Energy at Lawrence Livermore National Laboratory, is a mega-Joule class laser and target irradiation facility for investigating the physics associated with stockpile stewardship and inertial confinement fusion (ICF). It is designed to produce 1.8 MJ of frequency-tripled radiation (351 nm) in 192 independently focusable beams, each with an amplifier system that has a square clear aperture of approximately 40 cm x 40 cm. These amplifiers have aberrations that contribute to the net wavefront and affect both conversion efficiency of the frequency converters and the size of the focus spot on the target.

We have built an adaptive optics system to address static aberrations in the main beam line and precorrect dynamic aberrations induced in the main power amplifiers so that NIF can meet requirements for its conversion efficiency and focus spot size. The adaptive optics system comprises a deformable mirror, Hartmann sensor, reference source and controller electronics and is depicted in Figure 1.1 for a single beam line.

An adaptive optics system allows the overall laser system to be built using reasonable precision engineering tolerances, which helps to maintain costs, while still achieving its performance goals. Additionally, an adaptive optics system will allow for some steady-state system disturbances, such as optical mount settling, structural settling and thermal drift.

The work covered in this paper was completed between 1994 and 1997 as a development project. The primary goal was to develop a set of requirements and engineering solutions for the National Ignition Facility deformable mirror. The effort included designing, prototyping and testing a large aperture deformable mirror.

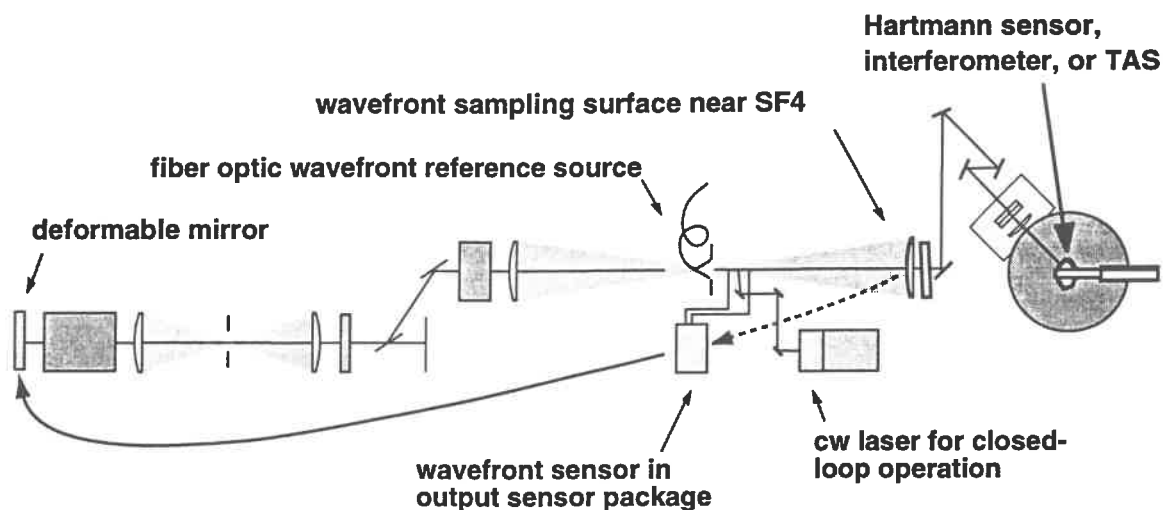


Figure 1.1 NIF adaptive optics system for single beam line.

2. Design Requirements

The design requirements established at the beginning of the project were more goal oriented, since the primary objective of this project was to establish requirements. Many of the requirements were set based on information gained from operating a small adaptive optics system on Beamlet.¹ Table 1.1 below overviews some of the key design requirements.

Parameter	Requirement
General	
Number of Actuators	39
Actuator Configuration	Hexagonal
Actuator Spacing	68.4 mm
Actuator Type	PMN
Clear Aperture	365 mm x 365 mm
Actuators	Replaceable on site
Residual Surface Error (closed loop)	0.05 waves RMS
Surface Correction Range	>4 waves P-V >1 wave P-V for 4 th order (@ 1.053)
Operating Environment	
Ambient Temperature	68°F+/-2°F
Relative Humidity	< 3%
Pressure	1 atm
Flash Lamp Radiation	10 J/cm ²
Mirror Bandwidth	
Open Loop (-3 dB)	>100 Hz

Table 1.1 NIF prototype deformable mirror design requirements.

Of the many design requirements, the residual surface error, flash lamp radiation and replaceable actuators combined to make the design extremely challenging. The residual

surface error was established to minimize the amount of uncorrectable error added to the laser system by the adaptive optics system, without significantly increasing the manufacturing cost. The requirement for flash lamp radiation is due to the location of the deformable mirror, only 2 meters away from the laser amplifier where Xenon flash lamps can produce up to 10 J/cm^2 with a 200 μsec pulse length. Lastly, the replaceable actuators requirement was established due to the lifetime of approximately 30 years and the desire to maintain the adaptive optics system at LLNL.

3. Design Description

The large aperture prototype deformable mirror consists of a 390 mm x 390 mm x 15 mm faceplate with 39 cylindrical 20 mm diameter posts forming equilateral triangular subapertures. The faceplate and posts are machined from a continuous piece of BK-7 glass. After fabrication, the front side is coated with $\text{HfO}_2/\text{SiO}_2$ and then stress compensated with a SiO_2 back side coating. The circular posts are attached, through flexures, to electrostrictive actuators made from lead magnesium niobate (PMN). Each actuator has an unloaded stroke of approximately 15 μm at 150 volts and has approximately 5 percent hysteresis. To protect epoxy used in connecting the posts to the flexures from the flash lamp energy, each circular post is coated with aluminum. The PMN actuator subassemblies are designed to be replaceable without removing the front faceplate. The actuator subassemblies are mechanically constrained in a 100 mm thick aluminum block. The coefficients of thermal expansion for aluminum and BK-7 are within 15 percent. Electrical leads pass out of the aluminum block to a connector. Figure 1.2 shows the fully assembled deformable mirror.

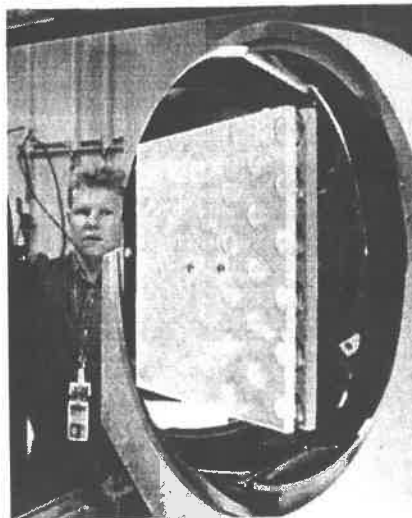


Figure 1.2 Large aperture 39 actuator prototype deformable mirror.

Designing the prototype deformable mirror utilized both experimental and modeling capabilities at LLNL. Finite element models were developed for both stress estimation and surface deformation trade studies. Experimental facilities such as the flash lamp laboratory and mechanical measurement laboratory were utilized to test concepts before building the 39 actuator deformable mirror. Furthermore, experimental testing was completed at both the coating vendor and actuator vendor facilities.

4. Laboratory and Beamlet Testing

Detailed testing utilized a 24" Wyko phase measuring interferometer in an off line laboratory before deploying the system on Beamlet. This testing included closed loop operation using a NIF prototype electronic controller and a prototype Hartmann sensor. Additional tests included aberration correction and influence function measurements. These results matched the expected design objectives and when combined with other off-line test results gave confidence in successful development of the system on Beamlet.

Further testing was completed on Beamlet, a single beam line prototype of NIF, to characterize performance under the actual environmental conditions. Several shots were completed which indicated no adverse prompt deformations of the surface of the deformable mirror when exposed to EMI or flash lamp energy, and no significant cleanliness degradation was observed. Furthermore, compared to operating a smaller deformable mirror at the laser input, the large intra-cavity deformable mirror was shown to reduce the focus spot size in the spatial filter pinholes in the beam line, as predicted. As expected, based on off-line tests, the output focal spot exceeded the NIF requirement which resulted in a design requirements change in the residual surface error.

5. Conclusion

A large aperture prototype deformable mirror was designed, fabricated and tested at Lawrence Livermore National Laboratory (LLNL) for future application to the National Ignition Facility (NIF). Experimental results, collected while operating the deformable mirror in a laboratory environment and on Beamlet, demonstrated the value of a large aperture deformable mirror, such as requiring less actuator stroke, due to the double-pass configuration, and producing a smaller focal spot size in the spatial filter pinholes. Lastly, this work proved that a large aperture adaptive optic system can be predictably designed, built and operated as an alternative to making a "perfect" optical system.

6. Acknowledgments

This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under contract W-7405-Eng-48.

7. References

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DESIGN OF MICRO-OPTICAL CONNECTOR AND NOVEL ROBOT HAND FOR PRECISE OPERATION IN A FIBER CROSS-CONNECT SYSTEM

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Abstract

A micro optical-connector and a robot hand have been designed for an optical fiber cross-connect system. The optical connector has a simple structure, enabling dense installation. The robot hand is compliantly supported, enabling reliable and automatic connection and disconnection. To determine the compliance value of the robot hand for reliable connection, we analyzed the connection process by using a static model.

Introduction

Optical networks, such as optical access networks for telecommunication services, are becoming widely used. One way to improve the flexibility and reliability of such networks is to use an optical-fiber cross-connect system. Such a system can make optical connections automatically; it can be used for restoring optical lines, reconfiguring optical networks, and reliably managing cabling systems. We are developing an optical-fiber cross-connect system for practical use¹⁾²⁾. It uses a newly designed micro optical connector that enables dense connector installation and a robot hand that operates reliably even when there is an error in its positioning. The result is a system that is small, reliable and low-cost.

Micro optical connector

The micro optical connector (Fig. 1) has the same basic structure as a standard MU-type optical connector. However, it is specially modified for robot handling. The connector plug consists of a ferrule with a diameter of 1.25 mm and a ferrule holder with a flange. The connector adaptor has a pair of plates to latch on to the optical plug. The adaptor also has a split alignment sleeve, a ferrule, and a coil spring inside. The split alignment sleeve forms 'C' shape and is elastic in its radius direction. After connection, The axes of the plug ferrule and adaptor ferrule are aligned with each other by the elastic force and physically contacted each other by the coil spring force of the sleeve. The result is a low-loss connection. The average connection loss for 75 connections was 0.15 dB, as shown in Fig. 2.

In the connection operation, the robot hand holds the connector plug and inserts it straight

into the connector adapter until the plates latch on to the plug's flange. The disconnection operation is illustrated in Fig. 3. The latch plates are tapered in the direction of the adaptor axes. The hand tips are tapered to fit the latch plates. The plug is removed from the adapter by the hand gripping the plug and drawing straight back from the adaptor. Because the connection and disconnection operations are done by simple motions of the robot hand, the structure of the connector can be simplified to achieve dense installation. Each connector occupies an area 31 mm^2 , whereas a standard MU connector occupies 50 mm^2 .

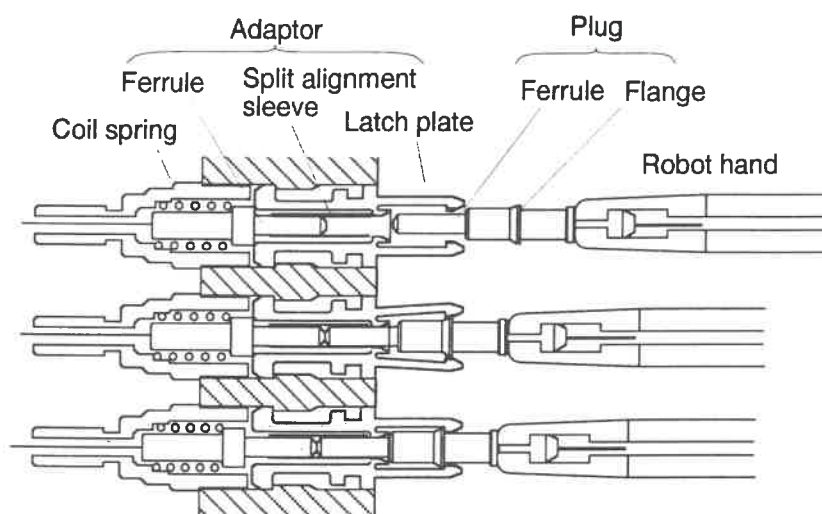


Fig. 1 Structure of micro optical-connector.

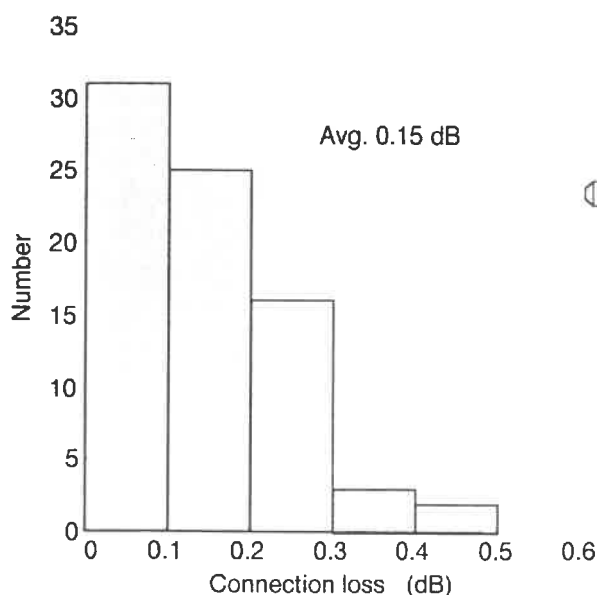


Fig. 2 Connection loss.

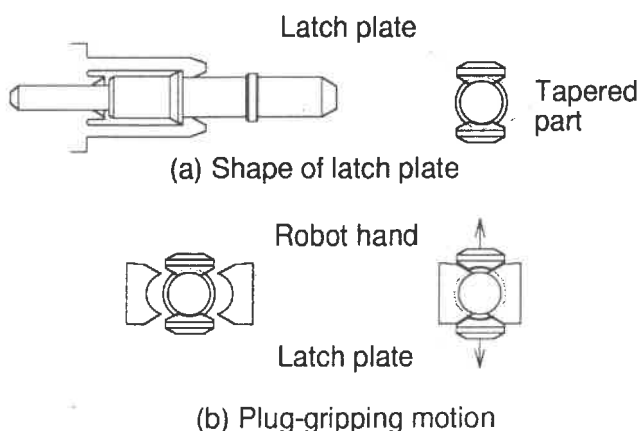


Fig. 3 Disconnecting operation

Robot hand

Errors in alignment between the plug and adaptor axes can occur during the connection operation due to a positioning error in the robot hand or an assembly error in the connector. To achieve reliable operation under such conditions, the robot hand has a compliance mechanism. As shown in Fig. 4, this mechanism consists of two parallel leaf mechanisms, giving the hand two degrees of freedom of compliance in the translational directions. The degrees of compliance can be adjusted by changing the leaf thickness.

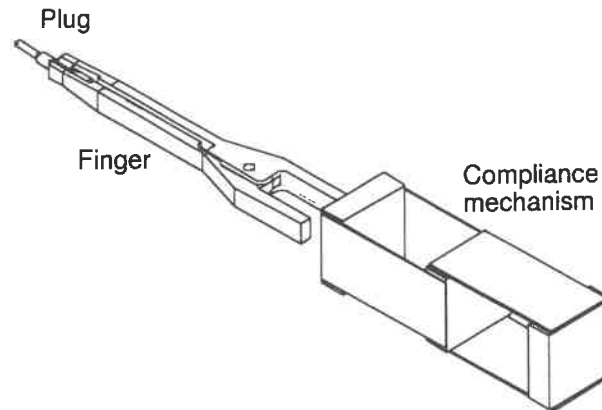


Fig. 4 Structure of robot hand.

Simulation and experiment

During the connection operation, the plug ferrule is inserted into the split alignment sleeve of the adaptor. Therefore, its modeling is similar that of the 'peg-and-hole' problem. However, jamming does not occur in this case because the split sleeve is elastic in its radius direction. Instead of jamming, the radius of the sleeve increases. Because the sleeve is made of ceramic, it can be break depending on the alignment error and the robot-hand compliance value. The insertion force required also depends on the compliance. An excessive force may be needed. To determine the compliance value for reliable connection, we analyzed the connection process by using a static model.

For our experiment, we used a simple adaptor structure: a sleeve and a ferrule. The adaptor ferrule was inserted into the split sleeve and held tightly. The plug was inserted into the sleeve, and the insertion force was measured. In the connection process, the contact status of the plug ferrule and the sleeve changes from one-point contact to two-point contact. The latter model is shown in Fig. 5. The plug ferrule is held by a compliant support with translational and angular stiffness. The sleeve is assumed to have a fixed center, and each side is assumed to be supported by stiffness. The insertion force was calculated taking into account the balance of contact forces in the model.

The simulated and experimental insertion force for $x_e=0.3$ mm and $\theta_e=1$ deg. as the initial positioning errors is shown in Fig. 6. A translational compliance of $k_x=0.25$ mm/N and angular compliance of $k_\theta=0.0017$ rad/N were used. The simulated force was almost same as the experimental one up to an insertion depth of about 2 mm. The difference is probably because the model is inexact when the angle of the ferrule to the sleeve is small.

Conclusion

Our proposed micro-optical connector has a simple structure enabling dense installation with a low loss (0.15 dB average). The proposed robot hand uses a mechanism with translational compliance, enabling reliable connection and disconnection. We are now enhancing our simulation model so as to clarify the relationship between the compliance value and the reliability. This connector and robot hand will enable the development of a practical optical-fiber cross-connect system.

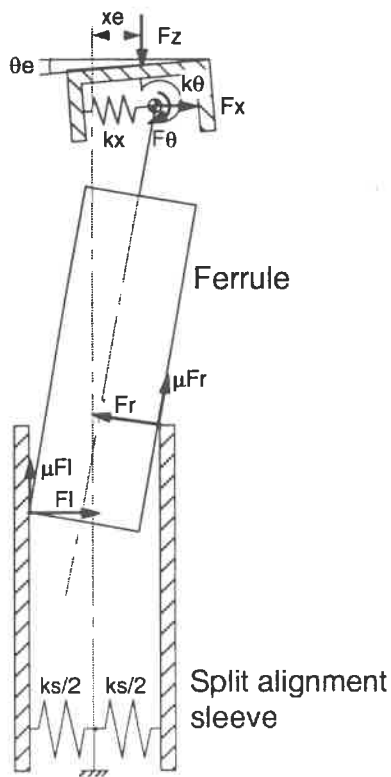


Fig. 5 Model of two-point contact.

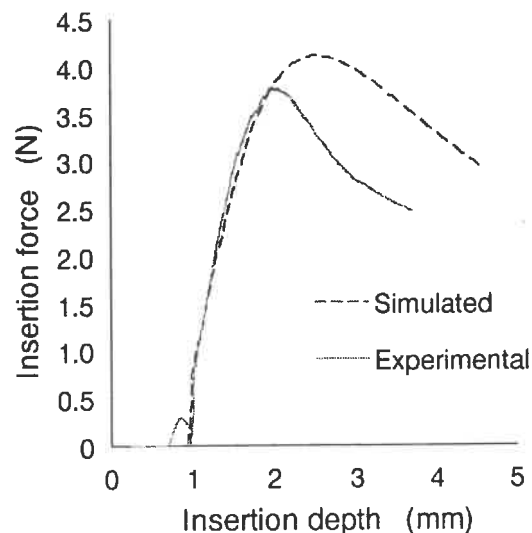


Fig. 6 Experimental and simulated insertion force.

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Neural network controllers for magnetostrictive actuator

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Abstract

In a general form, actual requirements as high performance and small sizes for mechatronic systems, has lead modern industry to design positioning systems with good characteristics of acceleration and positioning accuracy. By other way, the increasing demand of components with better metrological and finishing characteristics, as x-ray and infra-red lens, has allow the development of a number of types of micropositioning systems that are able to move machine elements in very small displacements with high level of accuracy. In this work it is proposed the use of a new type of actuator that applies the properties of eletromagnetic estriccion of certain metallic joining (magnetoestricative actuators). It is also proposed the application of digital control systems that uses control algorithms based on fuzzy logic (FL) and artificial neural networks (ANN) for the micropositioning control.

Keywords: precision engineering, micropositioning, magnetoestricative actuator, fuzzy logic, artificial neural networks.

1. Introduction

In precision engineering, adequate performance of machine tools depends upon the characteristics of the built-in positioning systems. The machining of complex shapes, with low dimensional and contour errors (of the order of nanometers), small depths of cut and optical grade surface texture are very difficult to accomplish with conventional machine tools. Most commercial precision machines do not have repeatability and accuracy capabilities to machine brittle materials (Blake & Scattergood, 1986). The development of positioning/correction systems of the tool is important as it is ultimately responsible for the form, finish and sub-surface integrity of a work-piece.

In this study, a modular type of tool holder which uses a magnetostrictive actuator is pro-

posed. However, magnetostrictive materials suffer from hysteresis type non-linearity, so that the output of such systems depends upon the previous input, and absolute positioning is only achievable with the aid of feedback control (Duduch. & Gee, 1990/ Rubio et al, 1996).

In this work, two control strategies based on artificial neural networks (ANN) are proposed for the positioning of a tool post in the nanometric range. In the first, a fuzzy logic (FL) look-up table mapping is proposed using a two layer neural network based on back-propagation with steepest-descent to cater for non-linearities (Freeman & Skapura, 1990). Thus, by using errors and error-rates in actuator motion direction, one can compute the increment of control input to the drive actuator.

In the second, a self-learning regulator is used which is based on Nguyen-Widrow emulator

proposition to aid the design of a suitable controller (1990). In this case, the Levenberg-Marquardt learning rule is used with back-propagation, to train the controller. Once trained the emulator plus controller network, this being done fixing the weight of the emulator, the neuro-controller can be used to drive the plant (Nguyen & Widrow, 1990).

The dynamic performance of magnetostrictive actuators are compared through numerical simulation to demonstrate the effectiveness of the proposed approaches and compared with results obtained by other authors

It is therefore necessary to use a micro servo tool positioning system which permits low amplitude movements and low following error to high frequencies ($< 10 \mu\text{m}$). One of the basic components of such systems is the actuator which changes an electrical signal into movement and is preferably based on solid state technology for direct action (Rubio & Duduch, 1998).

2. Micropositioning device and prototype

Most commercial precision machines do not have repeatability and accuracy capabilities to machine brittle materials. The development of positioning/correction systems of the tool is important as it is ultimately responsible for the form, finish and sub-surface integrity of a workpiece. Several devices have been proposed and developed. Figure 1 shows a modular type of tool holder which uses a flexure support system.

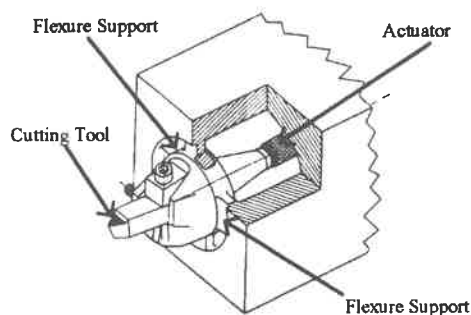


Fig. 1 - Modular Toolholder for Micropositioning

2.1 Magnetostrictive materials

The magnetostrictive materials transduce low voltage electrical energy into mechanical movement. The term Magnetostriction refers to the physical phenomenon of the dimensional variation that occurs in a magnetic material when a magnetic field is applied (Guillemin effect).

Originally developed as part of a US Navy project to find new sonar materials, ETREMA Terfenol-D[®] magnetostrictive actuators are built from an alloys of metals, terbium, dysprosium and iron. Figure 2 shows the physical properties of a magnetostrictive alloy Terfenol-D[®]. Among the characteristics that make this kind of actuator one of the best solutions, one can mention:

- High repeatability
- Reliability
- High load carrying capacity
- Large bandwidth
- scarcely affected by environmental changes

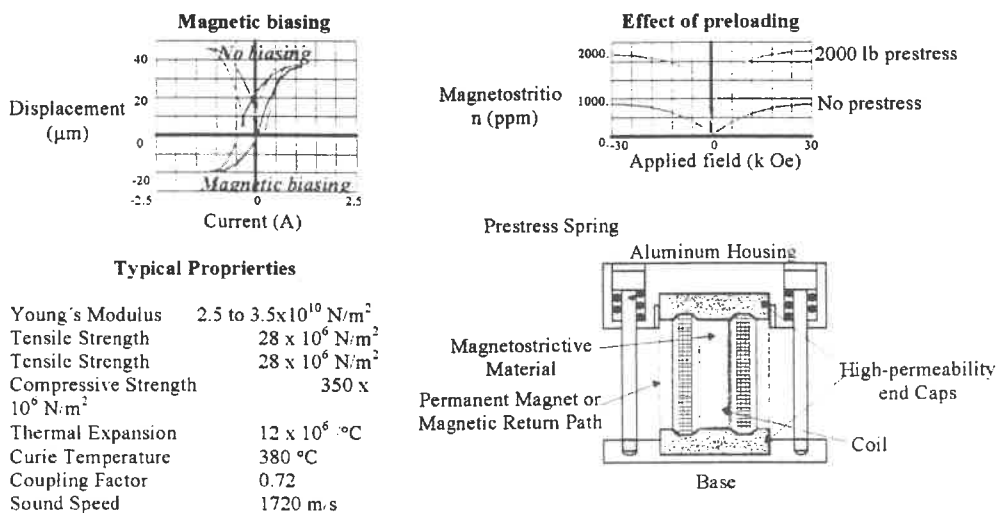


Fig. 2 - Magnetostriction (Goodfriend, 1994/Etrema Prod. Inc., 1994)

* ETREMA Products Inc: 2500 north loop drive. Ames, IA 50010

Its high performance makes this type of actuator an excellent alternative for micropositioners, vibration control, seismic sources, noise control.

3. Displacement control system

The next step is to propose a closed-loop position control system adequate for the system described above and able to meet (ultra-precision) required specifications. In this work, two control strategies based on artificial neural networks are proposed for tool post positioning in the nanometric range.

In the first, fuzzy logic (FL) lookup table mapping using a two layer neural network based on back-propagation with steepest-descent. will be considered.

In the second, a self-learning regulator is used which is based on Nguyen-Widrow emulator proposition to aid the design of a suitable controller (Nguyen-Widrow, 1990)

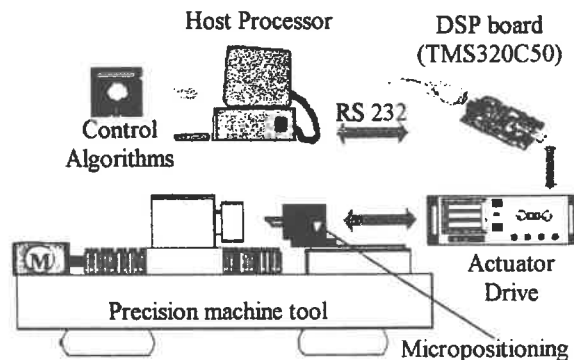


Fig. 3 - System configuration

4. - The control algorithms

An ANN with feed-forward architecture is a system composed of inputs, outputs and many simple and similar processing elements, all connected through internal parameters called weights (Freeman & Skapura, 1991). After specifying the number of layers, the number of neurons per layer and the transfer functions, it is necessary to train it, i.e., to adjust its weights, in order to achieve a desired input/output behavior. The training, in turn, can be accomplished using the error back-propagation algorithm, originally created for multi-layer networks and non-linear differentiable transfer functions (Freeman & Skapura, 1991). This method is normally based on

gradient descent where the parameters, such as weights and biases, are moved in the opposite direction to the error-gradient. It can reach a local, rather than a global, error minimum, for any function with a finite number of discontinuities. As presented by many authors, an ANN with one hidden layer consisting of sigmoidal neurons and one output layer with linear neurons, can map most non-linear function. The idea, as described by Nguyen and Widrow (1990), is that each neuron in the hidden layer can take a small part of the function relating its input to its output and make a linear approximation to that part. The output layer then adds the parts together to form the complete approximation to the desired function (Irie & Miyake, 1988).

Another learning process, also based on error back-propagation, utilizes the Levenberg-Marquardt approximation (see for example The MathWorks (1995) and Santoro et al (1998)). This optimization technique is more powerful and allows better performance through the Newton method. The Levenberg-Marquardt learning rule is given by:

$$\Delta W = (J^T J + \mu I)^{-1} J^T e \quad [01]$$

where J is the Jacobian matrix of the derivatives of each error for each weight, μ is a scalar and e is an error vector. If μ is small, equation (1) becomes the Gauss-Newton method which is faster and more accurate near the error minimum. Thus, the aim is to change μ in order to work either with the Gauss-Newton method or gradient descent, when the error is increased, μ is increased $J^T J$ becomes negligible. Learning takes place by using $\mu^{-1} J^T$ which is the gradient descent, otherwise, μ is decreased.

4.1 - Fuzzy logic look-up table mapping

Industrial applications of motion controllers based on fuzzy logic became popular due to their design facilities and also because it is not necessary to know the mathematical model of the plant. The design of a fuzzy controller consists of measuring some system variables, fuzzifying these quantities into fuzzy sets defined by the designer, check it with a knowledge based system (a set of rules of the type "if A then B") and return a crisp output by defuzzifying these quantities. Thus, for the controller, one has a look-up table with the error and rate of error as input and the control signal or its increment as out-

put. For intermediate values between those used in the universe of discourse, the normal procedure is to compute the output using interpolation. Trying to avoid either the look-up and interpolation processes based on mathematical equations, in this work the possibility is studied of mapping a fuzzy look-up table through an ANN with the Levenberg-Marquardt method.

The fuzzy "lock up" table corresponds to a matrix where the inputs are position errors and position-error variation. Thus, pair of input elements corresponds to an element of the matrix, whose value gives a quantitative measure of how is the system compared with the behaviour desired (Li and Lau, 1989).

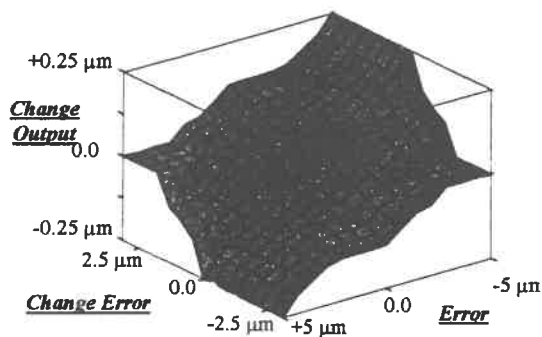


Fig. 4 - Rules Surface

4.2 Nguyen-Widrow proposition

The Nguyen-Widrow proposition [Nguyen & Widrow, 1990] is based on an emulator that helps the design of a robust controller. This emulator, a multi-layer ANN, learns to identify the system dynamics. The controller, another multi-layer ANN, learns then to control the emulator. Once is has trained the emulator plus controller network (being done fixing the weights of the emulator), the neuro-controller can be used to drive the actual plant. To train the emulator, the selected topology is shown in Figure 5-a, where it is assumed that the states of the plant are directly observable without noise. v are control inputs and E^k are the output variables at step k . A data base is generated from simulated data. To train the controller, the emulator is utilized in order to back-propagate the error through the network. This is justified since unfortunately, only the error at the output plant is available. Thus, for a two-layer neuro-controller, one has a network composed of four layers to be trained, as shown in Figure 6-b.

The objective is to train the ANN using the Levenberg-Marquardt learning method in order to achieve a set of weights that minimizes a performance index given by:

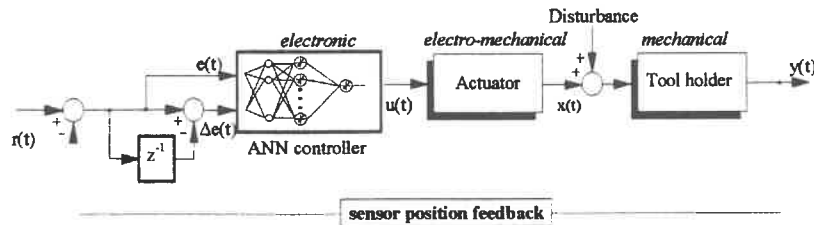


Fig. 5 - Displacement Control System

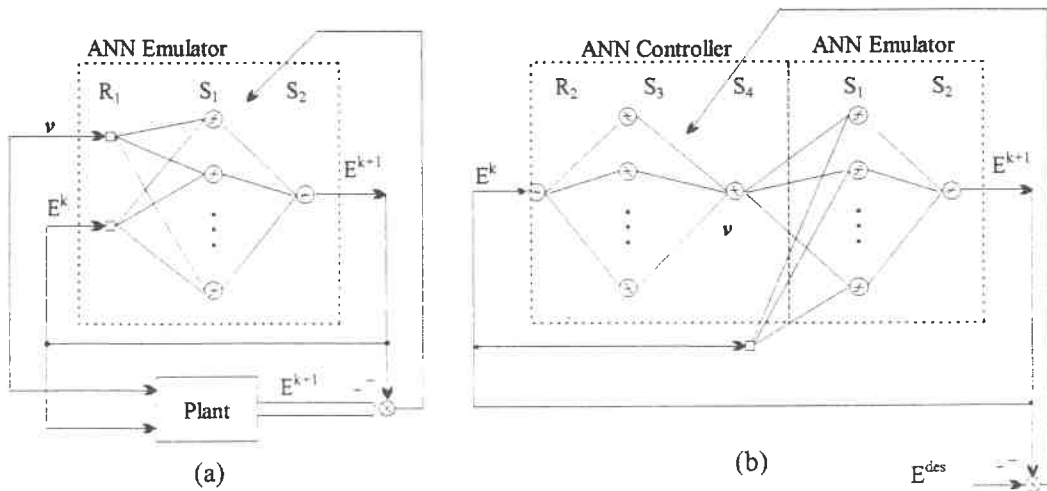


Fig. 6. -a) Emulator; b) Controller with the emulator

$$PI = E \left\| E_{x,y}^{des} - E_{x,y}^k \right\|^2 \quad [02]$$

where E is the mathematical expectation operator.

5. Modeling of micropositioning device

Each element of the dynamic system will be described by its respective mathematical model. This system may be divided into two parts: the digital controller and the plant which comprises the other elements (electro-mechanical components). On the plant, two parts can be clearly distinguished, one is the magnetostrictive actuator and the other is the moving mechanical elements (see Fig. 5).

Mechanical Elements (Tool holder)

$$\frac{Y(s)}{X(s)} = \frac{2 \cdot \xi \cdot \omega_c \cdot \left(s + \frac{\omega_c}{2 \cdot \xi} \right)}{s^2 + 2 \cdot \xi \cdot s + \omega_c^2} \quad [03]$$

Magnetostrictive Actuator;

$$\frac{X(s)}{I(s)} = \frac{K_{MST} \cdot \omega_n^2}{s^2 + 2 \cdot \xi \cdot s + \omega_n^2} \quad [04]$$

6. Numerical simulation

The parameter which will be used to assess the servo-controllers' behaviour is the transient response (as quick as possible, without overshoot and with little or no steady-state error).

The model proposed is a preliminary approximation and was obtained analytically (Rubio et al, 1996).

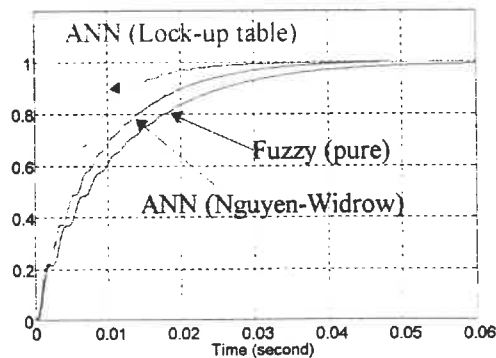


Fig. 7. - Transient response (unit input step)

Figure 7 show the closed-loop response obtained for unit input step excitation. The graphs were plotted through simulation of the dynamic system and controller via MATLAB® simulation software.

One can note that the dynamic behaviour of the plant was satisfactory since all responses obtained were fast and without overshoot. For the same type of input, time responses for the fuzzy-logic-based controller were very good. Only small steady-state error was noted. Changes in rules and membership functions may improve all responses.

Time responses for the neural controller using Levenberg-Marquadt were very fast and with a better performance in comparison with pure fuzzy-logic. Both fuzzy logic look-up table mapping and Nguyen-Widrow algorithm seem very satisfactory for the presented application.

7. Conclusions

This paper has presented the application of ANN on the design of two different controller for a magnetostrictive actuator-based micropositioning system. A fuzzy logic lookup table mapping using a two layer neural network and a self-learning regulator based on the Nguyen-Widrow proposition are proposed. The designed regulators appear to be very suitable and robust for attaining the objectives of fast response, low overshoot and low steady state error. Furthermore, the flexibility offered by ANN is efficient for the present application. The Levenberg-Marquadt method, shown to be powerful and quick when using back-propagation. The following advantages were: - it is not necessary to define a mathematical model of the plant and - processing time is smaller when compared with conventional controllers applied to similar systems.

This work is intended to show the development of a micropositioning system to be used in ultra-precision single-point diamond turning. The system utilises the micro-displacement characteristics of solid state actuators in the design of a micro-tool holder. The mathematical model proposed, although neglecting non-linearities such as static friction, variable stiffness and damping, represents fairly well the dynamic behaviour of the electro-mechanical linear system.

The dynamic system permits changing the actuator to suit the desired characteristics and

needs, making the system modular and easily re-configurable.

The comparison of the control strategies shown here allows insight of the system behaviour and judgement of modifications to be performed for improved performance.

Finally, this study point improved to the possibly of knowledge-based techniques, such as Fuzzy Logic and Neural Networks to control magnetostrictive actuators, as these techniques do not require detailed mathematical models to formulate the algorithms and have adaptive capability to intrinsic variations to the process.

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Performance Evaluation of Commercial Motion Control Boards

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The dynamic performance testing of two commercial motion control boards will be presented. This testing is targeted towards evaluation of commercial boards for use in non circular turning and to feedback results to the board manufactures for product improvements. If a commercial motion control board is found to have the dynamic range, then a costly controller development could be avoided. Currently, two commercial motion control boards are being evaluated on a high accuracy linear motion stage.

The testing criteria has been developed measures and qualifies the following attributes of a motion control board:

- dynamic resolution
- servo control algorithm accuracy and disturbance rejection
- block throughput rates
- trajectory generation accuracy and bandwidth

The dynamic resolution is the product of servo loop closure and the number of bits in the digital to analog conversion (DAC). With a fast closure rate and 14 to 16 bits of resolution on the DAC, a motion system is able to respond to disturbances and reject them. An example would be an interrupted cut in machining of non-circular workpieces. This process would subject the system to a sudden change in load or impulse. With an instrumented hammer, the test system is subjected to an impulse, the measured response (with an accelerometer) is the disturbance rejection in terms of stiffness and damping. At a minimum, 14 bits of DAC resolution are needed for a fast response to changing conditions.

In addition, the servo control algorithm used will affect the disturbance rejection. More sophisticated pole-placement algorithms perform better than PID algorithms. Since pole-placement algorithms are more computational intensive, the simple PID algorithms can be made to run quicker, this very fact will increase their performance. For all high speed motion applications, a velocity feedforward component is required in the servo algorithm.

How accurately a high speed contour/trajectory can be followed is dependent on block throughput rates, bandwidth and acceleration and deceleration look ahead algorithms. Each line of XYZ motion commands represents one block. For complex contours such as non circular turning, a large number of block are required to accurately machine the workpiece. A test was devised to measure block throughput by generating a sinusoidal motion program and measuring the frequency of motion with an accelerometer. As the number of blocks that represents the sinusoid increases to the limit of the block throughput, the frequency of the measured signal becomes distorted. Using the same technique, but changing the frequency of the motion program, the phase lag is measured and bandwidth determined.

In summary, the testing criteria and methods have been developed and will be presented along with the results of two commercial motion control boards evaluations.

POSITIONING ACCURACY OF SCARA-TYPE MANIPULATOR AND A METHOD OF ERROR COMPENSATION

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1. Introduction

SCARA-type manipulators are generally used in precision assembling systems. The requirements for positioning accuracy with the manipulators become more demanding as the assembled parts themselves become more precise. And nowadays, with the manipulators being used in CIM-systems, it is desirable that the positioning accuracy satisfy in the world coordinate system. However, existing manipulators have a positioning error of over 100s μm , and accuracy has not been easy to improved for the following reasons: a standard measuring method for the positioning accuracy of multi-joint manipulators has not been established, and the manipulators are controlled in semi-closed systems, so that their accuracy is affected by mechanical errors of the transmissions used in their joints¹⁾²⁾³⁾. This study will take up the measurement of the positioning accuracy of the SCARA-type manipulator to clarify the error sources, and then propose an error compensation method that can be verified experimentally.

2. Experimental set up and measurement principle

Fig. 1 shows the experimental set up. The manipulator used is a SCARA-type assembling robot. The upper arm length is $L_1=250$ mm, and the forearm length is $L_2=132$ mm, both nominal. Each joint is driven by a DC servo motor (100 W for joint θ_1 and 60 W for θ_2) and a harmonic-drive transmission (reduction ratio of 1/100 for θ_1 and 1/88 for θ_2). The rotational angle of each motor is sensed by a rotary encoder with resolution of 1000 P/R, and the motor rotation is controlled by detecting the encoder signal. With the arm pose almost fully expanded, the resolution of positioning is 24 μm .

The reference position in the world coordinate system is given by a reference board as shown in Fig. 2, and an XY stage. The board has 32 reference markers at intervals of 50 mm over a range of 350 $\times$ 150 mm. A reference position is defined as the intersection point of crossing lines of 10 μm width drawn on a marker. The

reference board has been calibrated on a 3-dimensional coordinate measuring machine to determine the actual position of the markers. The board is traversed by a precision X-Y stage in the range of 50 $\times$ 50 mm, so that the markers take any positions within their intervals. The error of the stage is less than 9 μm .

A CCD camera is attached to the end of the manipulator. It has 768 $\times$ 494 pixels, and its range of vision is 0.7 $\times$ 0.4 mm. When a positioning command for reference position is executed, the CCD takes the image of the cross line on the marker, and the positioning error is calculated. The calibration of the CCD is shown in Fig. 3: It has been proved that the measuring system has the resolution of 1.2 μm .

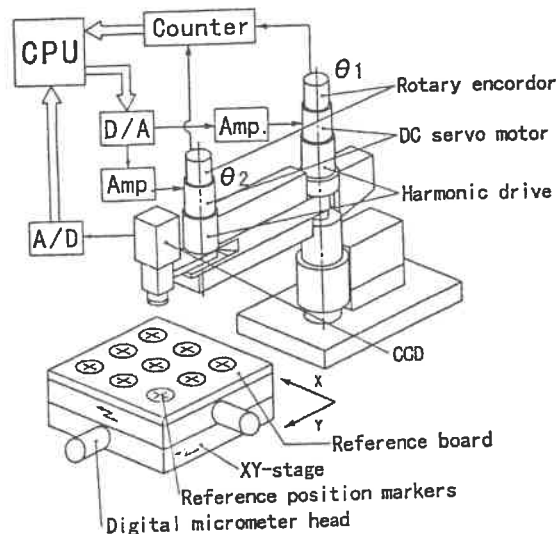


Fig. 1 Experimental set up

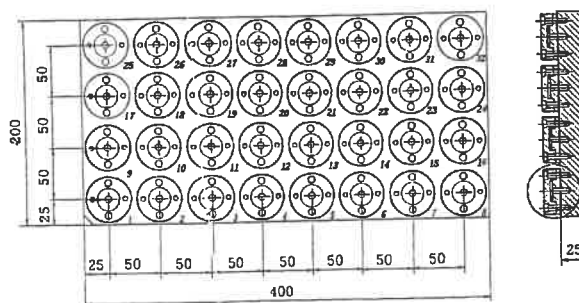


Fig. 2 Configuration of reference board

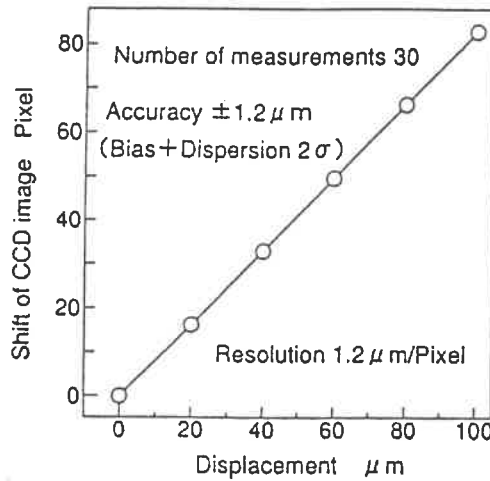


Fig. 3 Calibration of CCD

Fig. 4 shows the kinematics of the SCARA-type manipulator in the robot base coordinate system. The terminal position $E(x, y)$ is determined in the base coordinate system by the next equations:

$$\begin{cases} x = L_1 \cos \theta_1 + L_2 \cos(\theta_1 + \theta_2) \\ y = L_1 \sin \theta_1 + L_2 \sin(\theta_1 + \theta_2) \end{cases} \quad (1)$$

The accuracy of E is affected by the link parameters L_1 , L_2 , and the joint angles θ_1 , θ_2 . In the following experiments, positioning is executed and it confirms that the residual pulse of motor encoder is zero.

3. Measurement over global range at rough intervals

3.1 Measured result First, the XY stage is fixed, and the positioning errors are measured over the whole range of the reference board (350×150 mm) at intervals of 50 mm. The positioning to each reference is repeated 20 times under the identical conditions: Fig. 5 shows the measured result. In the figure, the arrows represent the bias error, while the ellipses represent the repeatability of $\pm 3\sigma$ (σ : standard deviation). The maximum bias error is $300 \mu\text{m}$, and the repeatability is $30 \mu\text{m}$ ($\pm 3\sigma$). The bias error is great, and errors of link parameters (L_1 , L_2 in eq. (1)) are considered the major cause of this bias.

3.2 Link parameter calibration Fig. 5 shows the position of any 3 points i , j and k defined in both the robot base coordinates (x, y) and the world coordinates (X, Y) . With the measurement mentioned above, the position of the robot terminal can be known accurately using the world coordinates determined by the position

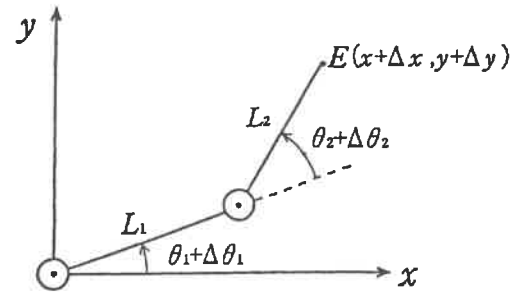


Fig. 4 Kinematics of SCARA-type manipulator

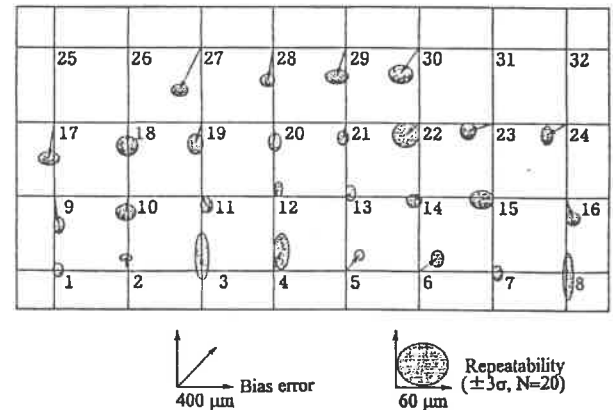


Fig. 5 Measured positioning errors over global range

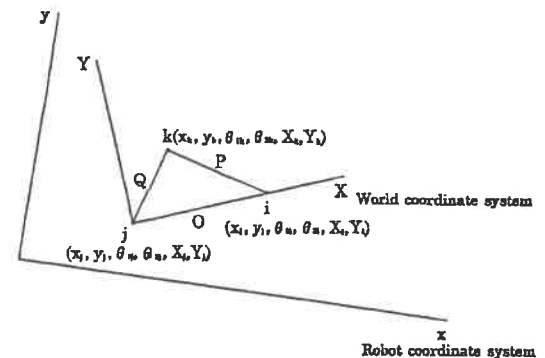


Fig. 6 Coordinates and joint angle for 3 points

of the reference markers and the positioning error measured by the CCD. On the other hand, the base coordinates can not be known accurately, since only the joint angles of θ_1 and θ_2 are known.

The base coordinates of any points $x_m = (x_m, y_m)$ ($m=i, j$, or k) are given by eq. (1) with joint angles θ_{1m} , θ_{2m} , and link parameters L_1 , L_2 . Because the distance between any 2 points of i, j , and k is constant, the next equations can be formulated.

$$\|x_j - x_i\|^2 = O^2, \quad \|x_k - x_j\|^2 = P^2, \quad \|x_i - x_k\|^2 = Q^2 \quad (2)$$

The distance O , P or Q is calculated using world

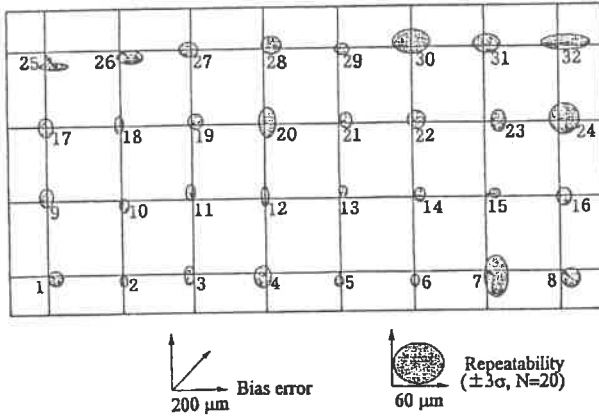


Fig. 7 Positioning error after link parameter calibration

coordinates. Substituting eq. (1) into eq. (2), L_1 and L_2 are given by the next equations:

$$\begin{cases} L_1 = \sqrt{\frac{d(O^2S_{jk} - P^2S_{ij}) - b(O^2S_{ki} - Q^2S_{ij})}{ad - bc}} \\ L_2 = \sqrt{\frac{a(O^2S_{ki} - Q^2S_{ij}) - c(O^2S_{jk} - P^2S_{ij})}{ad - bc}} \end{cases} \quad (3)$$

The coefficients in eq. (3) are as follows.

$$\begin{aligned} a &= R_{ij}S_{jk} - S_{ij}R_{jk}, \quad b = T_{ij}S_{jk} - S_{ij}T_{jk} \\ c &= R_{ij}S_{ki} - S_{ij}R_{ki}, \quad d = T_{ij}S_{ki} - S_{ij}T_{ki} \\ R_{vw} &= \{(\cos\theta_{1v} - \cos\theta_{1w})^2 + (\sin\theta_{1v} - \sin\theta_{1w})^2\} \\ S_{vw} &= 2\{(\cos\theta_{1v} - \cos\theta_{1w})\{\cos(\theta_{1v} + \theta_{2v}) - \cos(\theta_{1w} + \theta_{2w})\} \\ &\quad + (\sin\theta_{1v} - \sin\theta_{1w})\{\sin(\theta_{1v} + \theta_{2v}) - \sin(\theta_{1w} + \theta_{2w})\}\} \\ T_{vw} &= \{\cos(\theta_{1v} + \theta_{2v}) - \cos(\theta_{1w} + \theta_{2w})\}^2 \\ &\quad + \{\sin(\theta_{1v} + \theta_{2v}) - \sin(\theta_{1w} + \theta_{2w})\}^2 \end{aligned} \quad (4)$$

In eq. (4), v and w mean $(v, w) = (i, j), (j, k)$ or (k, i) .

Because 32 points are given on the reference board, there are ${}_{32}C_3 = 4960$ different combinations of the necessary 3 points, i.e. 4960 different pairs of link parameters are obtained from the measured results. Their mean values are $L_1 = 250.015$ mm and $L_2 = 131.450$ mm, and they are considered the most probable values. Using the obtained link parameters, positioning is executed again. Fig. 7 shows the positioning errors after this calibration: The bias error has been reduced to less than $70 \mu\text{m}$.

4. Measurement in local range at fine intervals

4.1 Positioning error Using the XY stage to carry the reference board, it is possible to measure the positioning error within an interval of 50 mm between the reference markers: Fig. 8 shows an example of the

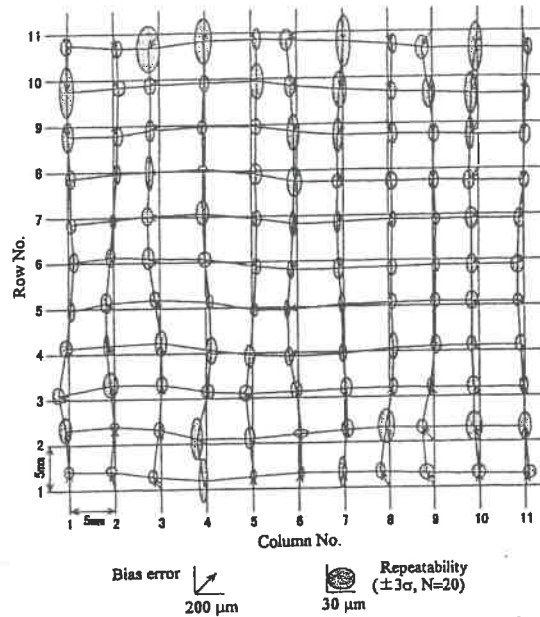


Fig. 8 Example of positioning errors within local range

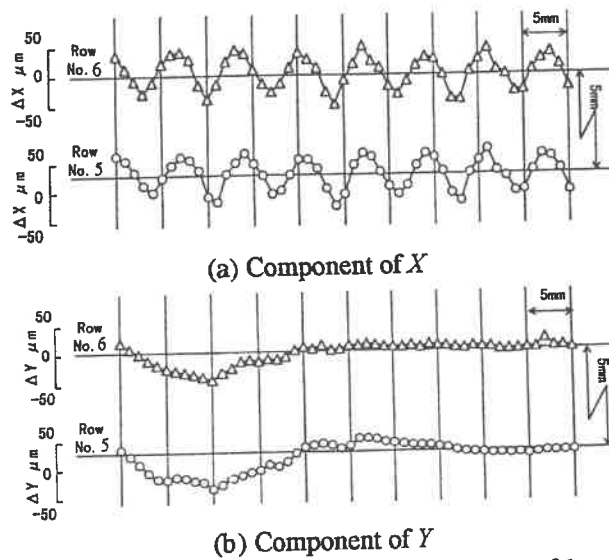
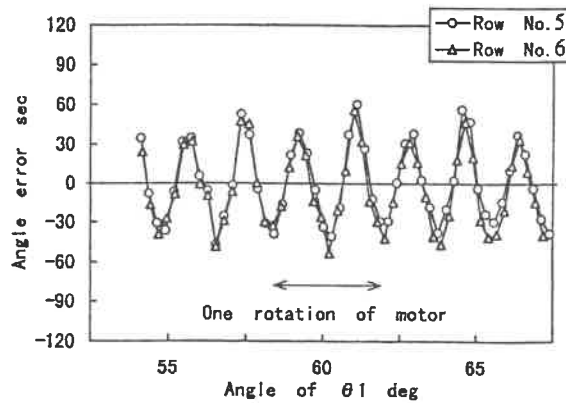


Fig. 9 Positioning errors measured at intervals of 1 mm

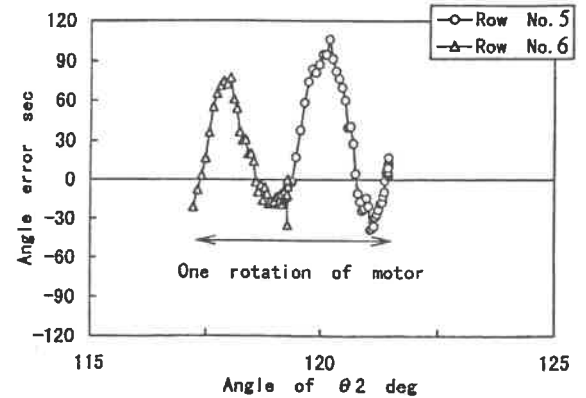
measured result. The positioning target is No. 12 marker on the board, and the marker is moved by the XY stage with an interval of 5 mm. The result in Fig. 8 was obtained before the link parameter calibration.

After the link parameter calibration, the positioning errors are measured at finer intervals. Fig. 9 shows the bias errors of the positioning along rows No. 5 and 6 at an interval of 1 mm. In the figure, the errors are described by composites of Δx and Δy , and the errors have periodic characteristics.

4.2 Conversion into joint angle error When the joint angles θ_1 and θ_2 have errors of $\Delta\theta_1$ and $\Delta\theta_2$, the positioning errors caused by the propagation of the



(a) Error of θ_1



(b) Error of θ_2

Fig. 10

Joint angle errors

angle errors are given as the following equations derived from eq. (1).

$$\begin{cases} \Delta x = \frac{\partial x}{\partial \theta_1} \cdot \Delta \theta_1 + \frac{\partial x}{\partial \theta_2} \cdot \Delta \theta_2 \\ \Delta y = \frac{\partial y}{\partial \theta_1} \cdot \Delta \theta_1 + \frac{\partial y}{\partial \theta_2} \cdot \Delta \theta_2 \end{cases} \quad (5)$$

Solving eq. (5) inversely for $\Delta \theta_1$ and $\Delta \theta_2$, the joint angle errors can be obtained from the measured positioning errors Δx and Δy .

Fig. 10 shows the joint angle errors converted from the positioning errors of Fig. 9: It is clear that the joint angle errors are periodic. Moreover, the period of the error corresponds to a half rotation of each motor. On the basis of these results, it can be concluded that a positioning error within local range is caused mostly by the periodic error of the harmonic drive transmission.

5. Error compensation

Since the periodic errors of the transmissions are repeatable, it is possible to compensate for the error. The periodic errors of the joint angles in Fig. 10 are approximated by sinusoidal functions, which are stored in the controller, and the positioning command is compensated for by these functions.

Fig. 11 shows the effect of the compensation: The bias errors amount to $85 \mu\text{m}$ without the compensation, whereas they are reduced to less than $24 \mu\text{m}$ with the compensation. The value matches the positioning resolution determined by the motor encoder, and the availability of the method is thereby verified.

6. Conclusions

- (1) Using the reference board carried on the XY

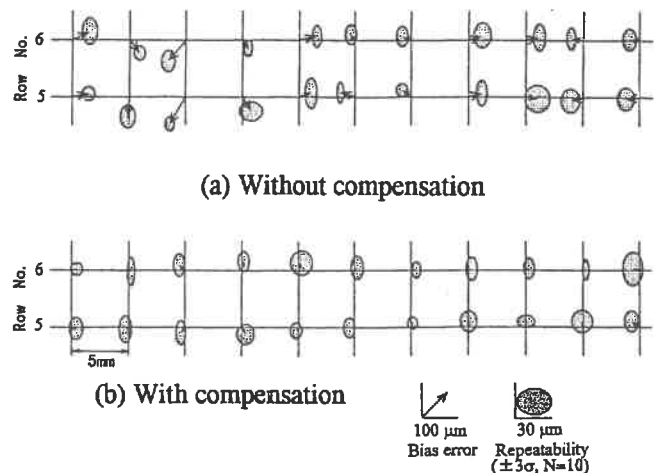


Fig. 11 Effect of compensation on positioning errors

stage and a CCD camera, the measurement system for the positioning accuracy of SCARA-type manipulators is constructed.

- (2) From measured results over a global range, the link parameters are calibrated.
- (3) Considering the measured results within local range, it is clear that the positioning errors are caused mostly by the periodic error of the harmonic drive transmission.
- (4) A simple method of compensating for the periodic errors is proposed, and the availability is verified experimentally.

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Condition Monitoring of Stacked Piezoelectric Actuator Using Induced Charge

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1. Introduction

Because piezoelectric elements (piezo) have a high response and are small in size, they are used for positioning devices in various fields [1]. A major problem is the hysteresis of the displacement to the applied voltage in the use of piezos. The hysteresis causes positioning error or control instability of systems. Many methods are proposed to reduce or to analyze the hysteresis of piezos [2]-[6]. Newcomb et al reported that the displacement of a piezo is proportional to the supplied charge in the charge control [3]. However, the high output impedance of a current source restricts the response of the system. Piezos should be driven with a voltage source with low output impedance to obtain high response. The authors have proposed a displacement control method for piezos by using induced charge [7][8]. The piezo is driven with a voltage source and additional devices for the measurement do not restrict the size of an actuator in this method.

This paper describes the condition monitoring of the piezos by using the induced charge. At first, the principle of condition monitoring is introduced. Then the response of the induced charge is compared with that of the displacement of a piezo. Finally,

displacement control is demonstrated by using induced charge.

2. Principle of condition monitoring

Fig. 1 illustrates the principle of the displacement measurement by using induced charges [7][8]. A multilayer piezo consists of stacked thin piezo plates with depositing electrodes. Two sets of electrodes are wired in parallel. Conductive plates (the electrodes for the detection) are attached on the end surfaces of the piezo. When the voltage is applied to the piezo, charges which are proportional to the internal charge are induced on the electrode for the detection. The induced charge can be measured with detectors as a voltage meter or a charge meter with high input impedance. The internal charge indicates the displacement of the piezo [3], and the induced charge is proportional to the internal charge. Therefore, the induced charge can be also used to estimate the displacement of the piezo.

3. Measurement of response of induced charge

3.1 Experimental setup

A multilayer piezo with the dimensions of $10 \times 10 \times 20$ mm is used in the experiments. It extends $16 \mu\text{m}$ at the applied voltage of 150 V. Electrodes for the detection are attached on both end surfaces of the piezo. The upper electrode for the detection is made of aluminum with $15 \times 15 \times 1$ mm in size and 0.7 g in

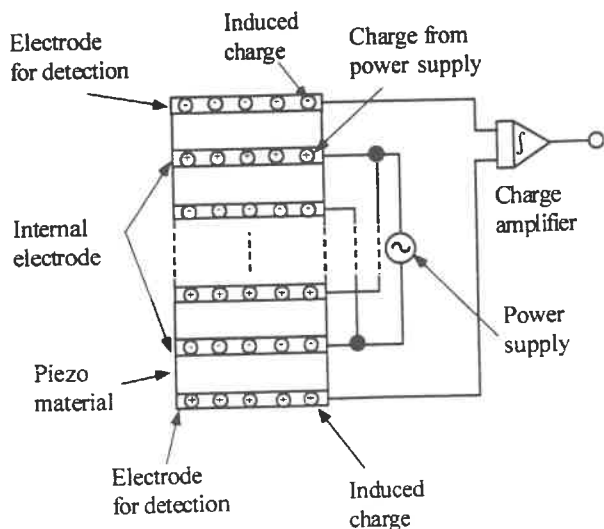


Fig. 1 Principle of displacement measurement using induced charge

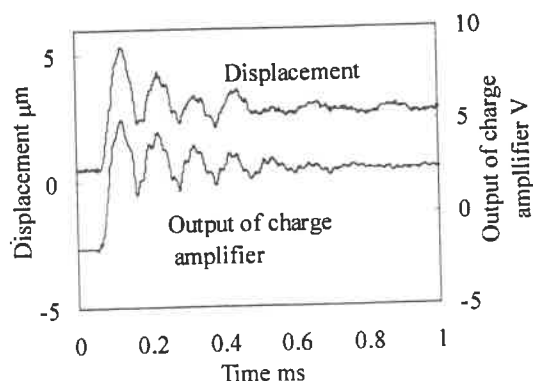
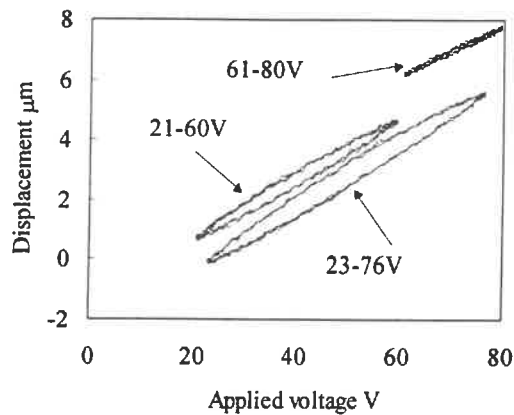
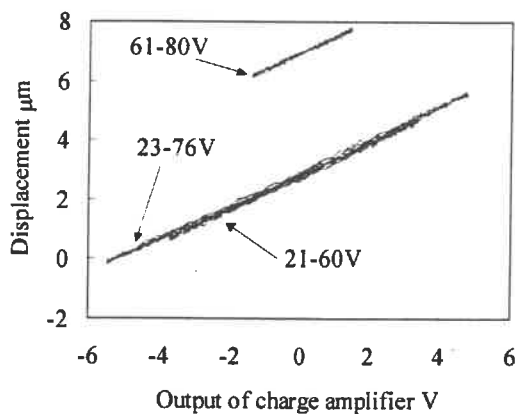


Fig. 2 Transient response of piezo



(a) Displacement vs. applied voltage



(b) Displacement vs. induced charge

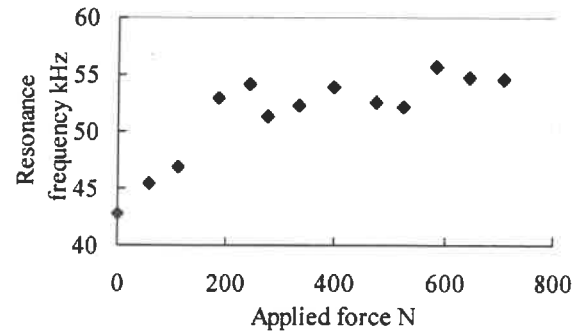
Fig. 3 Hysteresis loop

weight. The lower electrode for the detection fixed on the grounded table is made of carbon steel. The electric wires are connected with the electrodes for the detection by conductive glue. The displacement of the piezo is measured with a fiber-optic displacement sensor with the sensitivity of $0.011 \mu\text{m}/\text{mV}$ and the frequency range up to 100 kHz. The induced charge is measured with a charge amplifier with the frequency range from 2 Hz to 200 kHz. The coupling capacitance between the electrodes for the detection and the piezo is 50 pF.

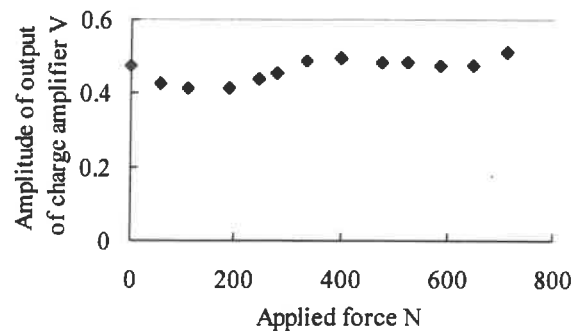
3.2 Response of induced charge

Fig. 2 shows step response of displacement and induced charge [7]. A step response of the displacement of the piezo is the same as a step response of the output of the charge amplifier. The correlation coefficient is 0.951 between them.

Fig. 3 shows examples of the hysteresis loop of the piezo [7]. The piezo is driven at the frequency of 2.5 kHz. Because the induced charge is measured in steady state, the average output of the charge amplifier is zero. No hysteresis of the displacement is observed to the



(a) Resonance frequency vs. applied force



(b) Amplitude of induced charge vs. applied force

Fig. 4 Relationship between induced charge and applied force

output of the charge amplifier. A difference of the ratio of the displacement to the applied voltage is 48 % between the large amplitude and the small amplitude. The ratio to the output of the charge amplifier is 0.9 %. Induced charge is approximately 200 pC at the applied voltage of 25 V.

Static external force is applied to the piezo by screwing a bolt. Fig. 4 shows the relationship between resonance frequency and the applied static force. Sinusoidal wave with the amplitude of 1 V and the DC bias voltage of 50 V are applied to the piezo while sweeping the frequency. The applied force is measured with a load cell with strain gauges. As the static force increases, the resonance frequency becomes higher. However, the amplitude at the resonance frequency is almost constant. Therefore, the external force applied to the piezo can be estimated by measuring a resonance frequency of the system.

4. Application to displacement control of piezo

Fig. 5 illustrates a system configuration of displacement control. The piezo and the charge amplifier is the same ones as described in Sec. 3. A personal computer (80386, 16 MHz) is used for a controller. A low pass filter with the cutoff frequency of 50 Hz precedes the piezo driver in cascade because the shortest sampling time of the computer is 0.3 ms.

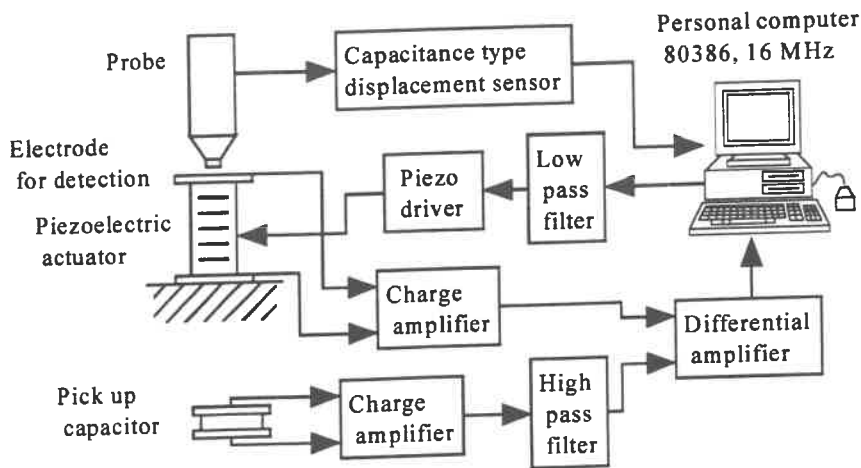


Fig. 5 System configuration

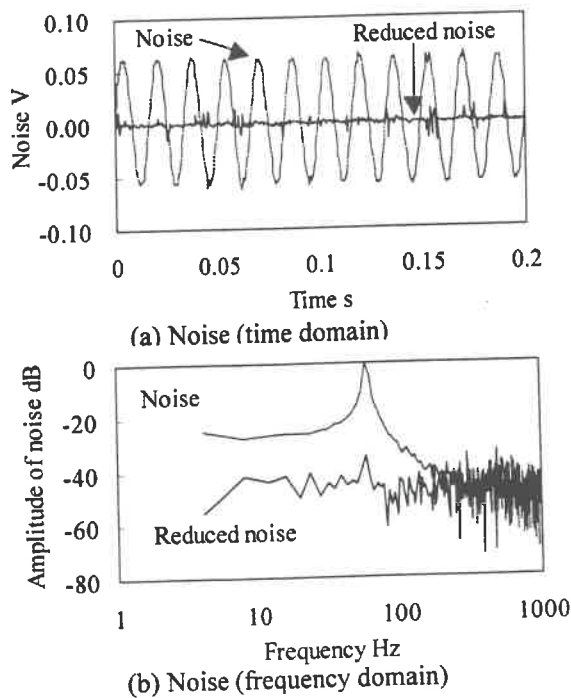


Fig. 6 Active noise reduction

The displacement of the piezo is measured with a capacitance displacement sensor with the resolution of 5 nm and the frequency range of 40 kHz.

The signal from an additional capacitor to pick up noise is subtracted from the signal from the charge amplifier to reduce the noise from power lines. The phase of the signal from the pick up capacitor is adjusted by using a high pass filter. Fig. 6 shows output signals before and after noise reduction. The noise from power lines is reduced approximately 40 dB.

The DC component of the output of the charge amplifier is neglected because it has the same frequency characteristics as a high pass filter.

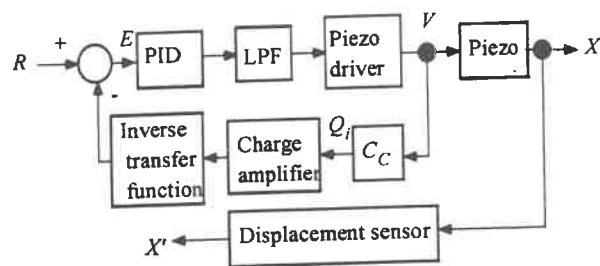


Fig. 7 Block diagram of systems

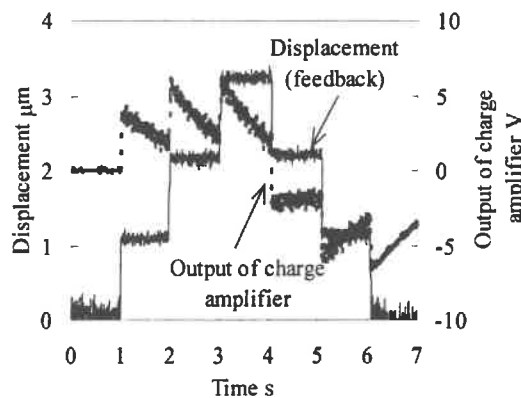
Therefore, it is compensated by using an inverse transfer function [9]. Fig. 7 illustrates a block diagram of displacement control system with inverse transfer function compensation [8]. Because the transfer function of the charge amplifier is represented by

$$G(s) = \frac{V_o(s)}{Q(s)} = \frac{s^2}{s^2 + 2\zeta\omega_c s + \omega_c^2} \quad (1)$$

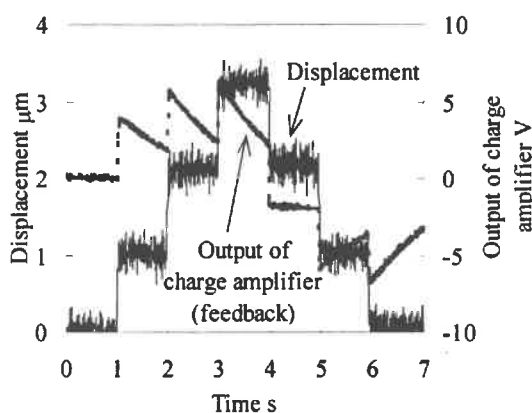
the output of the compensator v_{comp} is represented by

$$\begin{aligned} v_{comp}(n) = & 2v_{comp}(n-1) - v_{comp}(n-2) \\ & + (1 + 2\zeta\omega_c T + \omega_c^2 T^2)v_o(n) \\ & - (2 + 2\zeta\omega_c)v_o(n-1) + v_o(n-2) \end{aligned} \quad (2)$$

where ζ is the damping ratio, ω_c is the resonance frequency, T is the sampling time, v_o is the raw output of charge amplifier, and n is clock. Fig. 8 shows examples of positioning by proportional control. The reference displacement is changed by 1 μm at every 1 s. The output of the charge amplifier is decreased as shown in Fig. 8 because the lower cutoff frequency is 2 Hz. However, steplike displacement can be obtained by the induced charge feedback control with the inverse transfer function compensation as well as by the displacement feedback control. The displacement of the piezo can be controlled for a longer time than the time determined by the cutoff frequency of the charge amplifier by the inverse



(a) Displacement feedback control



(b) Induced charge feedback control with compensation

Fig. 8 Examples of positioning

transfer function compensation.

The positioning performance by the induced charge feedback control is compared with open loop control and displacement feedback control in case of step response. Table 1 shows the positioning accuracy by PID control. The average and standard deviation of the settling time by the induced feedback control is the same as that by the displacement feedback control.

5. Conclusions

The positioning performance by the induced charge feedback control is compared with that by displacement feedback control. The displacement of the piezo can be controlled by the induced charge feedback control as well as by displacement feedback control. The authors will apply this method to actual positioning devices using piezos.

Acknowledgment

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Table 1 Comparison of displacement control

	Steady state error μm		Settling time ms	
	μ	σ	μ	σ
Open loop control	-0.01	0.05	11.9	1.5
Displacement FB	-0.04	0.04	2.8	0.0
Induced charge FB+ITFC	0.05	0.15	2.8	0.1

μ : average, σ : standard deviation, FB: feedback, ITFC: inverse transfer function compensation

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Linear Frequency Response Testing of Microservos Using (Non-linear) Multifringe Interferometric Position Measurement

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1 Interferometers: Capability and History

Sub light-wave capabilities of the optical interferometer have been exploited by the optical physics community for fine motion measurement over more than a century since Michelson investigated (putative) solar orbital velocity effects on the velocity of light. He later used the same instrumental arrangement to calibrate lead-screws for grating ruling. With the increasing need for sub-micrometre capability to measure and control the performance of ultra-precision instruments and machines, the capabilities of the interferometer have extended to contemporary precision manufacture.

A major constraint was originally motion-range, (limited by source-coherence and fringe-visibility). By the 1950s, the recently-available 198 isotope of mercury, used in an RF-excited lamp, yielded coherence length and uninterrupted measurement/control over a range of the order of 100mm about the central equal-path position. Slightly extended, this enabled 250mm gratings to be ruled under interferometric control (Harrison & Archer 1951, Harrison & Stroke 1955). Since then, the continuous wave gas laser has extended fringe visibility to tens of metres (Dukes & Gordon 1970). Interferometric control of grating-ruling went on to become the technology used for VLSI microchip wafer-stepper positioning.

2 Displacement Measurement Modes

The interferometer produces an intensity fringe-pattern whose non-linear repetitive incremental nature tends to favour multiple fringe-count ranges with a basic incremental resolution of one half wavelength ($\sim 300\text{nm}$ using the helium-neon gas-laser 632.8nm transition spectral line). An alternative measurement mode is to limit the range to the pseudo-linear region of about 1/3 fringe range centred on the mid-intensity value. This may seem unduly restrictive but it was employed over much larger ranges by phase-comparing the interferometer output with a reference signal derived from the mechanical motion system, with respect to which it is to be controlled, and to synchronise the two with a servo working within the permissible small sub-fringe error range (Harrison & Stroke 1955, Dyson *et al* 1970, Dyson 1970).

Because the interferometer generates a fringe pattern displacement repeated for each half-wavelength mirror-displacement, by the machine control engineer, its operation may thus be seen as offering two distinct modes. In fringe counting mode, the range/resolution ratio will exceed 1 part in 10^6 for ranges $> \sim 0.3\text{m}$ with the instrument used solely as a digital transducer.

High resolution/range values can be obtained over lower displacement ranges by interpolation between fringes. However, as the range comes down to of the order of a few fringes, the issue of transition between incremental and continuous analogue modes becomes increasingly significant, requiring a combination of dedicated switched and analogue signal-handling.

3 Frequency Response Testing

In frequency-response testing of position servos, the amplitude and phase of the output motion is required to be measured and compared respectively with those of an input driving sinusoid over a range of frequencies representing the system bandwidth.

Where the motion is to be measured interferometrically, over a range of the order of a few micrometres to nanometric resolution, a compound count and interpolative arrangement is normally required. This must convert the repeated (non-linear) sine-squared fringe intensity function to a continuous linear measure of rectilinear displacement using the nearest fringe-transitions as local datum start-points. Fringe counting can be combined with table 'look-up' to convert the repeated (non-linear) sine-squared fringe intensity function to a continuous, linearly proportional measure of rectilinear displacement.

4 Modification of Lissajou Figure Technique

An input $A\sin\omega t$ to a linear system will result in an output signal $B\sin(\omega t + \phi)$ where the system gain or attenuation is B/A and its output phase lag is ϕ rads. Lissajou's arrangement offers ready means of comparison of one signal with another at the same synchronous frequency.

When used in conjunction with linear measures of the input and output signals, comparison is readily made by displaying the input and output signals as respectively X and Y spot deflections on an oscilloscope and the resulting observed trace will generally be an ellipse with inclined axes. The special case of an inclined straight line indicates negligible phase lag and of a non-inclined ellipse indicates 90° lag between signals of different amplitudes; (a circle additionally indicating balanced amplitudes). In practise, rather than analyse the observed trace, it is usual to employ a dual-output signal generator using the zero phase output as the actuator drive, the signal generator adjustable-phase output as the 'scope X-input and the actuator position-transducer signal as the Y-input. The ellipse can then be nulled to a straight line by adjusting the phase-control and the phase-difference read from the signal generator phase setting. Attenuation/gain levels are usually resolved with reference to the transfer characteristics of the input interface and the motion-transducer voltage/displacement sensitivity.

The foregoing requires an output transducer whose voltage/displacement transfer characteristic is mathematically *linear*. (This term is intended to describe the transfer function characteristic for a range of amplitudes. Common usage of the same term when referring simply to rectilinear motion may lead to ambiguity. It should be understood that here we are measuring the *rectilinear* motion of an actuator using a transducer manifesting a highly *non-linear* voltage/displacement characteristic). Being highly non-linear, the interferometer-based voltage/displacement transducer would not normally be considered compatible with the Lissajou technique. However, it was noted that the interferometer output is nevertheless cyclically harmonic (complementary with the nature of the test signals) and it is this property which permits it to be employed effectively in Lissajou

system frequency-response testing. Using the interferometer, motion is represented by cyclic fringe-passage (rather than by 'scope-spot Y displacement). With this change of operational sense, the cyclic interferometer output signal may be compared and analysed harmonically with respect to a sinusoidal input driver signal.

As before, the test-reference and output signals are arranged respectively to be X and Y oscilloscope inputs. For open-loop testing of a position servo actuator by interferometric measurement, fringe-passage corresponding to motion amplitude will be observed as repeatedly reversed multifringe cycles (of sine-squared form when photodetected with linear voltage/intensity sensitivity) between stationary points determined by the interferometer and amplitude settings.

In contrast to nulling an ellipse to a straight line when using a linear output transducer, in the case of the interferometer signal, when it has negligible phase lag between input signal and output motion, the fringe signal will be in-phase with the reference signal set at zero phase-lag. In this condition, a reversing cyclic fringe-trace of *constant spatial period* will be prescribed by the 'scope spot due to reference signal sinusoidal time-base speed changes being in-phase with those of the actuator motion generating the fringe-signal. If however the actuator sinusoidal motion lags the reference-signal, respective speed variations will be dephased and variations will be observed in the spatial periodicity of the fringe-pattern appearing on the 'scope. Adjustment of the reference signal phase will achieve constant spacing at the null setting, whence the phase lag can be read from the reference signal phase control setting of the signal generator, as in the traditional (linear) Lissajou Method.

5 Instrumental Construction and Operation

A two-beam Michelson/Twyman-Green interferometer was constructed with arm length of 15mm and mounted contiguously with the frame of a fast tool servo actuator unit from an ultra-precision single-point diamond turning machine so as to measure the displacement of a small mirror mounted on the tool-holder. Two standard photovoltaic silicon photodiodes in a bridge circuit, set at a quarter-fringe spacing, measured the fringe-displacement and the light source was a 633nm 1mW laser beam (chopper-wheel amplitude modulated at 2.4kHz). Phase-sensitive detection was employed using an E.G&G. Lock-in amplifier. However, satisfactory results were obtained at base-band frequencies without resort to noise reduction.

It is necessary that mechanical and wavelength stability of the instrumental set-up should permit sufficient repeatable oscillatory conditions for observing and adjusting phase-nulling. In a single-storey former Metrology Lab. at Cranfield with earth-banked walls and 2m deep concrete floor, elementary anti-vibration precautions were required to ensure this (e.g. avoidance of mechanical cantilever overhangs in instrumental mechanical design). Observations were made with the actuator and interferometer mounted on a bench-mounted surface plate. Elementary baffling was employed to reduce barometric-noise from air-currents. In respect of drift, remarkably, the end-point phase of the oscillatory fringe-pattern did not appear to shift appreciably during large fractions of an hour.

6 Advantages of the Modified Lissajou Approach

This adaptation of the Lissajou method has the considerable benefit of permitting servo-actuator responses to be observed and measured over ranges of a few μm (i.e. a small number of fringes) which do not lend themselves well to either continuous or incremental measurement. It permits

measurement and investigation of hysteretic signals suffering lost motion on reversal which may or may not be a function of amplitude (Duduch and Gee 1990, Slocum, 1992).

7 Conclusions

A method has been proposed and demonstrated which circumnavigates the non-linearity of the optical interferometer to achieve linear analysis by exploiting the harmonic nature of its output in a novel modification of the Lissajou technique. Using it, the operation of microservomechanisms may readily be observed and analysed over ranges of the order of 0.5-50 μ m without resort to dedicated equipment to take account of fringe-transitions and counts.

8 Acknowledgments

The authors thank Prof. P.A.McKeown and Cranfield Precision for use of the fast tool servo.

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Displacement to Frictional Load Characteristics of Plain Bearing Guideway Systems and Motion Analysis of NC Machine Tools

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1. Introduction

As previously reported⁽¹⁾⁽²⁾, the displacement to frictional load characteristics of plain bearing guideway systems is divided into two distinguishing regions or a transient region and a stationary one. The former is a transient travel range of several hundred nanometer to the stationary frictional load after reversing the feed direction of slide and in this range the frictional load shows the equivalent spring-like load up to several Newton per nanometer in stiffness. The latter is a stationary region where the frictional load is almost constant to be zero stiffness. The transient region gives dominant effect on the transient phenomena in minute stepwise motion of the slide, and it is known that there exists a characteristic performance of feed motion of the position control system applying plain bearing guideways, which is a new field to be analyzed. Positioning and contouring accuracy of NC controlled mechanism generated by summation of minute stepwise motion are seriously suffered from the characteristic performance of feed motion caused by the transient phenomena in the displacement to frictional load characteristics.

2. Construction of test rig

The test rig is shown in Fig. 1. The slide to be position-controlled is 450 x 300 mm in dimension, 150 kg in mass, supported vertically by plain bearing guideways and horizontally hydrostatically. It is driven by the force-operated hydraulic actuator and the feedback sensor as a displacement detector is of a capacitive type. The positioning control system was constructed as a continuous-data control one for avoiding the effect of discrete-data control on the transient characteristics. PI compensation was applied. The materials of the guideways and the slide are FC250 and FC300 respectively. Lubrication system is of a oil bath. The viscosity grade of lubricant is ISO #46. The test rig as a ultraprecision positioning mechanism has a feed resolution of 1 nm⁽³⁾⁽⁴⁾.

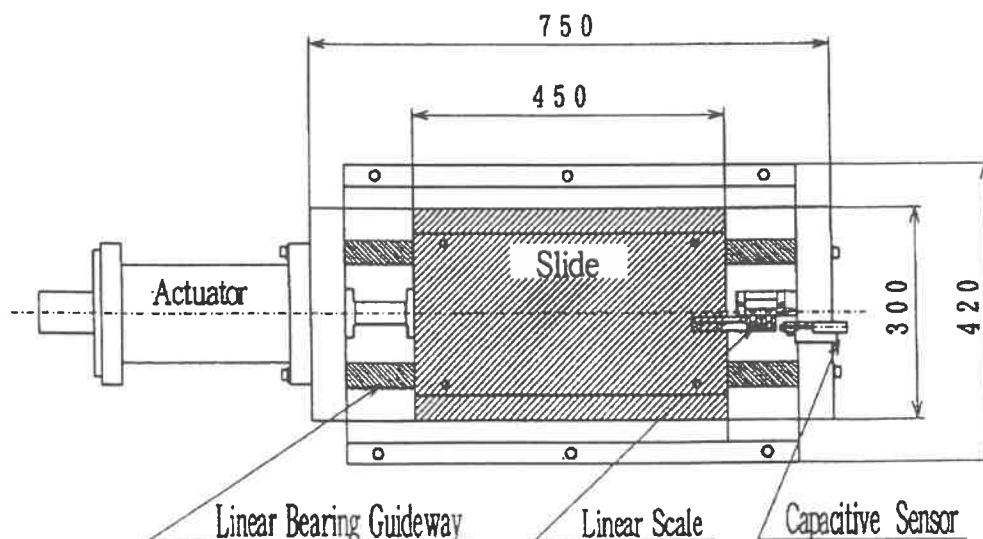


Fig. 1 Test rig

3. Displacement to frictional load characteristics of plain bearing guideways

Fig. 2 shows the displacement to frictional load characteristics after reversing feed direction, and (a) is a case of reversing negative to positive direction and (b) is from positive to negative in direction. The displacement on the abscissa is the cumulative displacement after reversing feed direction. In both cases there is a transient region of about 300 nm in displacement from the stationary friction before reversing feed direction to the stationary one after reversing. The equivalent spring constant derived from the displacement to frictional load characteristic curve is also shown in the figure. The equivalent spring shows a soft-spring characteristics. The equivalent spring constant just after reversing feed direction is 3.6 N/nm and the one after 100 nm displacement from the reversing position is 1.6 N/nm. The spring constant in the stationary region of the friction is zero and then the load characteristics of the positioning servo system has a great difference between the cases in the transient region and the stationary region. The transient phenomena in minute stepwise motion suffers greatly influence from this load characteristics.

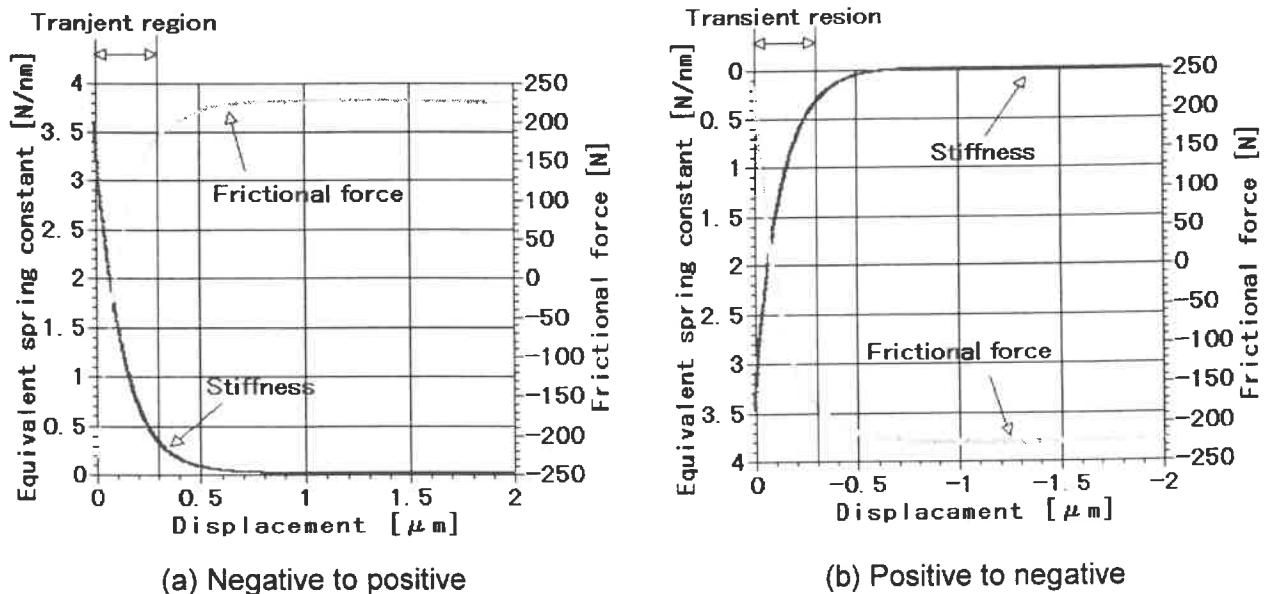


Fig. 2 Displacement to frictional load characteristics

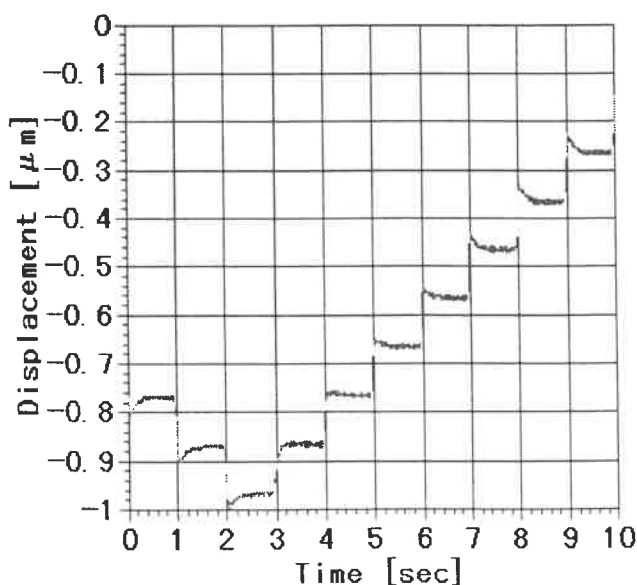


Fig. 3 100 nm/step feed

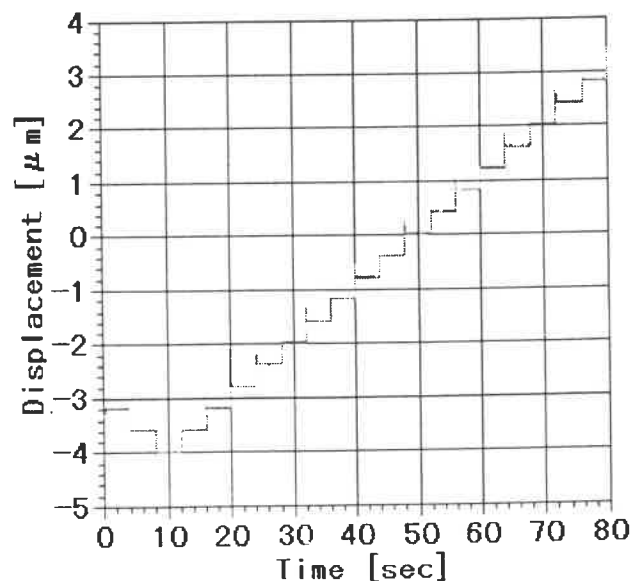


Fig. 4 400 nm/step feed

4. Transient positioning characteristics in minute stepwise feed motion

Fig. 3 and 4 are the responses to stepwise feeds of 100 nm/step, which is within the transient travel range, and of 400 nm/step, which is over the transient travel range, respectively. In a case of 100 nm/step feed, there is no overshoot for three steps after reversing feed direction, however in the fourth and following steps the overshoot occurs. In a case of 400 nm/step feed, the overshoot occurs in every stepwise feed, because of the unit step involving the high equivalent stiffness in the transient region and also approximately zero stiffness in the stationary region.

Fig 5 is summaries of the overshoots to the stepwise feeds shown in Fig. 3 and 4. The step number shows the sequential number of the steps after reversing feed direction. Fig. 5(a) shows the cases of the size of step being less than the transient travel range such as 50, 100 and 200 nm/step. Fig. 5(b) is for the size of step being over the transient travel range. The overshoot is defined to be percentage to the size of step. In Fig. 5(a), there is no overshoot as far as the cumulative displacement is within the transient travel range and the overshoot occurs over it. The displacement characteristics is completely different depending on the cumulative displacement being within or over the transient travel range. It comes from the difference of load characteristics to the servo system in the two cases. That is, within the transient travel range giving great spring-like load to the servo system the system is stable. However, in the stationary range giving approximately zero spring load, the massive load is predominant and a stability of the system is reduced. In the case of the size of step over 400 nm in Fig. 5(b), there occurs alternatively the overshoot more than 50 % for every size of steps. Such complicated phenomena in stepwise feed mechanism have not been widely observed.

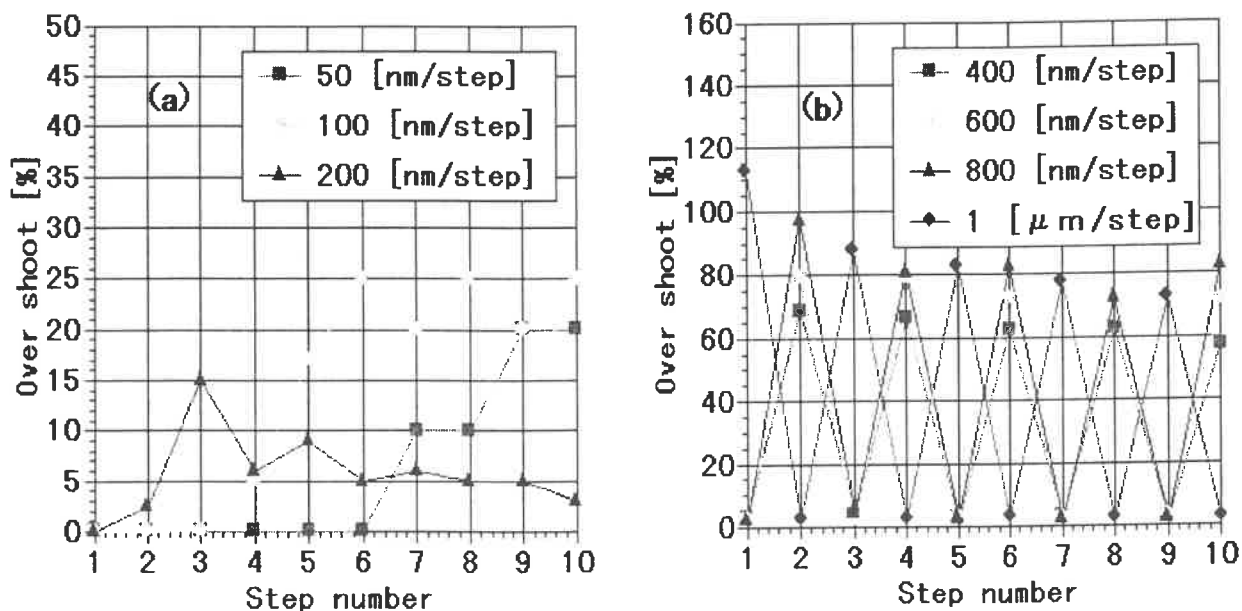


Fig. 5 Overshoot in stepwise feed

Fig. 6(a) and (b) show the displacement for 400 nm/step feed from 2nd step through 6th step and the corresponding frictional load respectively. The circle mark in Fig. 2(a) means a rising point. The frictional load at the rising point of stepwise feed is about +130 N for the 2nd step with great overshoot and -130 N for the 3rd step with small overshoot. From the rising point 2 the displacement starts along the displacement to frictional load characteristic curve from a point in the transient region with small spring-like stiffness near to the stationary region toward the stationary region with zero spring-like stiffness in Fig. 2(a). Then, the great overshoot occurs and for compensating the overshoot the system generates the reversal driving force along the curve in Fig. 2(b) to -130 N at the final position. The 3rd step starts under the frictional load

of “-130 N”. The equivalent spring constant at the rising point is about 2.5 N/nm as shown in Fig. 2(a) and then the positioning of the 3rd step involves the wide transient travel range. It depends on the transient characteristics of every step such as the overshoot, the damping performance of the system and the residual frictional load of the previous step which point is the rising point of step on the displacement to frictional load characteristic curve and also decides the transient performance of stepwise motion. The displacement characteristics shown in Fig. 6 is a typical example resulted from the displacement to frictional load characteristics in shown in Fig. 2, which is a new field of microtribology of a plain bearing guideway system.

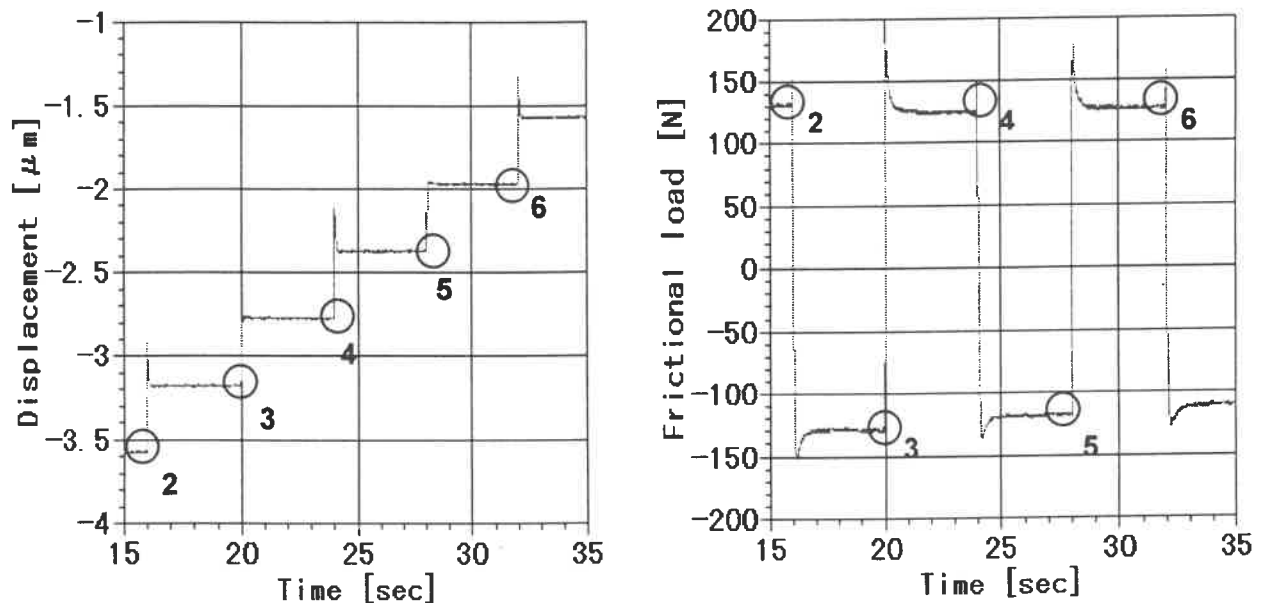


Fig. 6 Displacement and Frictional Load in 400 nm/step feed

5. Conclusion

The transient phenomena in minute stepwise feed of positioning servo system on plain bearing guideways are complicated, which have not ever been known. And these performances can be analytically explained by applying the displacement to frictional load characteristic curve of plain bearing guideways. This kind of performances can dominate a dynamic motion accuracy of NC controlled servo system and is very critical to be considered.

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Ultra precision positioning system for servo motor-piezo actuator using the dual servo loop and digital filter implementation

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1. Introduction

In recent years, the ultra positioning technique has become one of the important part in the development of precision engineering. In this paper, an ultra precision positioning system has been developed using a dual servo loop control. For a positioning system to obtain long distance with ultra precision, the combination of global stage and micro stage was required. A servo motor based ball screw is used in the global stage and the piezo actuator in the micro stage. For the improvement of positional precision, the digital Cheby Shev filter is implemented in the developed dual servo system. Therefore, the positional repeatability has been achieved within nano meter, and this technique can be applied to develop precision semiconductor equipment such as lithography steppers and prober.

2. System Design

The designed ultra precision positioning system is composed of a global stage and micro stage. The global stage is a combination of an X-Y stage and a servo motor based ball screw. The global stage plays part in relatively long distance positioning. The micro stage is a combination of a flexure hinged stage and PZT. The micro stage plays part in ultra positioning upon the global stage. Sampling time of the system is fixed at 10kHz by using a hardware interrupt routine.

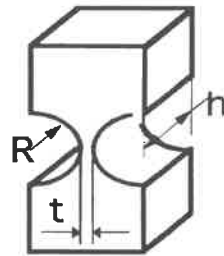
2.1 Global stage design

The designed global stage can move over 40 cm distances. The implemented ball screw for the moving stage has accuracy over $5\text{ }\mu\text{m}$. The global stage is controlled by a DSP(Digital Signal Processing) board with encoder feedback. The DSP board is installed to a PC and controls the servo motor using PID(proportional integral derivative) control.

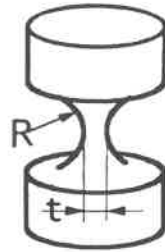
2.2 Micro stage design

For ultra precision positioning, the micro stage has been designed using a flexure hinge. The ultra precision moving mechanism using a flexure hinge has no friction, backlash and thus it has a

continuous, smooth action. One of the important points ,when designing a flexure hinge, is the bending stiffness. By Paros etal[1] and Smith etal[2] , bending stiffness of notched hinge and circular hinge shown at Fig.1 are as follows, where R is radius of notch, t is thickness of notch, h is width of notch, in case that $2R+t = b$.



Noched Hinge



Circular Hinge

$$K_B = \frac{2Eht^{5/2}}{9\pi R^{1/2}} \quad (\text{notched hinge}) \quad (1)$$

$$K_B = \frac{Et^{7/2}}{20R^{1/2}} \quad (\text{circular hinge}) \quad (2)$$

Fig.1 Typical flexure hinges : notched flexure, circular flexure

Other static and dynamic factors for designing the micro stage based at (1), (2) are as follows. q is displacement in the micro stage and L is effective distance from mid point of stage to flexure hinge.

$$\text{Total elastic energy of micro stage } U = K_B \frac{q^2}{L^2} \quad (3)$$

$$\text{Force toward q direction } F_q = \frac{\delta U}{\delta q} = 2K_B \frac{q}{L^2} \quad (4)$$

$$\text{Stiffness toward q direction } \lambda_q = \frac{F_q}{q} = \frac{2K_B}{L^2} = \frac{4Ebt^{5/2}}{9\pi L^2 R^{1/2}} \quad (5)$$

$$\text{Natural frequency of micro stage } \omega_n = \sqrt{\frac{4Ebt^{2/5}}{9\pi R^{1/2} L^2 M + 12\pi R^{1/2} ml^2}} \quad (6)$$

Therefor, the micro stage was designed.

2.3 Digital filter design

The digital Cheby Shev filter, which is to improve performance of the position control, has been implemented in the dual servo system. The measurement system used in the dual servo system has a vibration of magnetic chuck and bar which add a noise signal to the measured signal. So, frequency of the noise signal is to be cut-off, using a digital filter. This filter has cut off bandwidth frequency of 40~55Hz, ripple of 0.3, and 10th order, and it was designed at MATLAB etal[3]. The performance of position control is compared at Fig.2 and Fig.3

3. Dual servo loop control system implemented with digital filter

3.1 Configuration of dual servo system

A block diagram of the dual servo loop for the positioning system is shown at Fig.7. PZT is used as an actuator for the micro stage, and the servo motor and ball screw is used as an actuator for the global stage. System position is fed back by laser interferometer to the PC.

3.2 Algorithm of dual servo loop control

The flow chart of the dual servo loop control is shown at Fig.6. When the system target position is input, the global stage is operated for system position error to be within $10\text{ }\mu\text{m}$. After global stage positioning is performed, the micro stage is operated to reduce system position error precisely. The global stage is controlled by a PID control loop in the DSP board, and the micro stage uses an IP(Integral Proportional) control loop with a 10 kHz sampling time by hardware interrupt routine in the PC.

3.3 Dual servo loop control

The diagram of the dual servo loop is shown at Fig.8. For verifying the effectiveness of the developed dual servo loop control positioning system, $100\text{ }\mu\text{m}$ target position was input to dual servo system. At first, when using only the global stage without micro stage, response of $100\text{ }\mu\text{m}$ command is shown at Fig.4. There has been about $3\text{ }\mu\text{m}$ systematic error with noise signal. But using the dual servo loop system, shown at Fig.5, very little systematic error is found with noise signal less than 10nm due to digital filter implementation.

4. Conclusion

- (1) The dual servo loop control system, which is consist of servo motor(global stage) and PZT(micro stage), has been developed to move over 200mm distance with positional accuracy of 10 nano meter .
- (2) The micro stage was designed, and the static and dynamic performance of stage was evaluated.
- (3) For improving positional accuracy, frequency of vibration components was analyzed and digital filter has been implemented, giving better positional accuracy.

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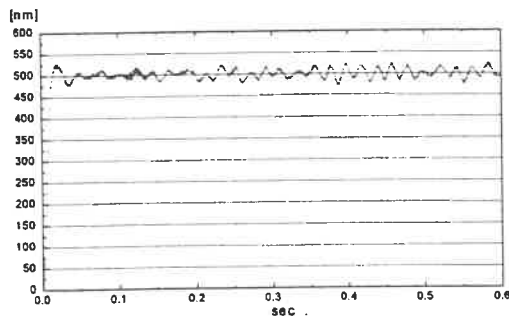


Fig. 2 500nm step response without Cheby Shev filter

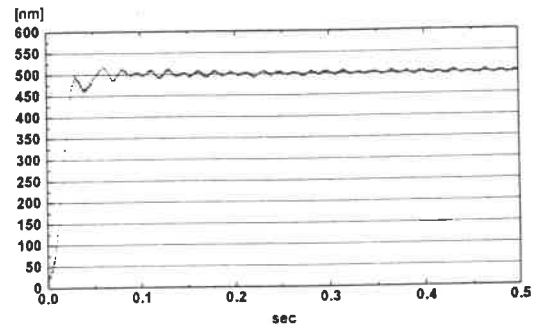


Fig. 3 500nm step response with Cheby Shev filter

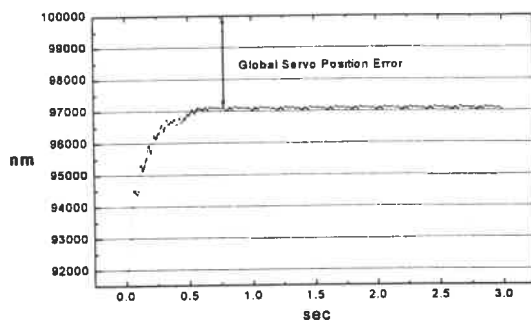


Fig. 4 Response of global stage under 100um command showing 3um positional error with noise components

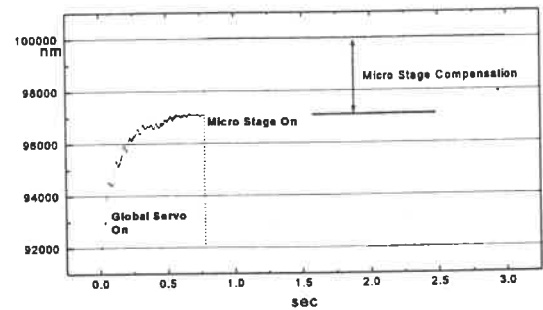


Fig. 5 Response of global-micro stage under 100um command showing about 10 nm noise components

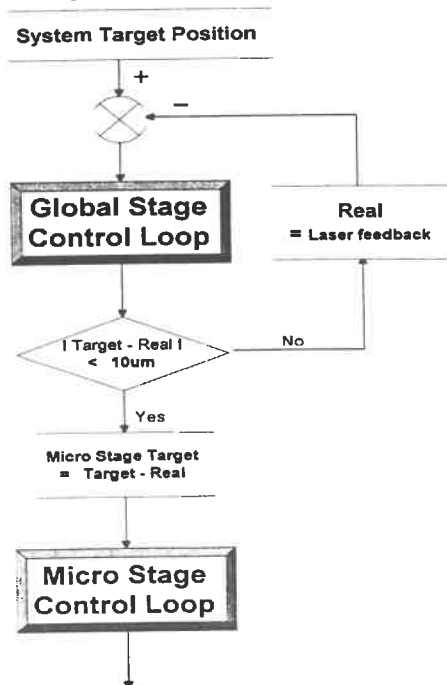


Fig. 6 Flow chart for dual servo loop

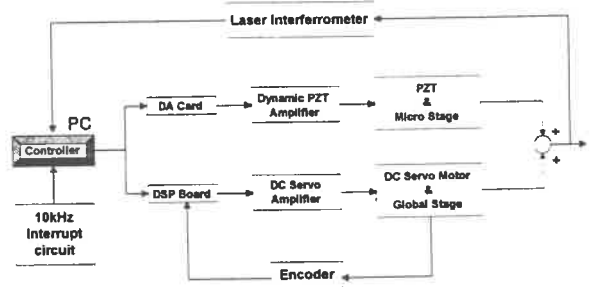


Fig. 7 Block diagram of the dual servo loop for positioning system

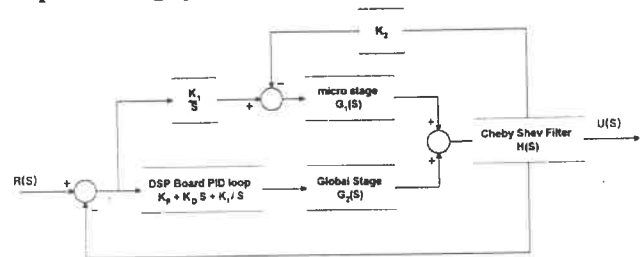


Fig. 8 Block diagram for the dual servo loop implementing digital filter

Development of a thermal error measurement/correction system for the 5 DOF spindle thermal deformations in CNC machine tools

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Introduction

Improvement of machine tool accuracy is an essential part of quality control in manufacturing process. Because of continuous demand for precision machined parts, various methods are applied. These method have ability of reducing or correcting the errors of machined parts. There are numerous sources of error in machine tools, such as geometric errors of the structrual elements, kinematic errors related to motion of these elements, static and dynamic errors with loading/unloading workpiece, cutting related error, and thermally induced error. Among these errors, the thermally induced error is reported as much as 70% of total errors. So many reserchers are trying to reduce or compensate the thermal error. In this paper, software thermal error measurement/correction system is developed and tested, where three different thermal error model is implemented. As result, thermal error can be reduced to 20~30% of total thermal errors..

Thermal error model : Multiple Linear Regression

This method is to use the linear algorithm for thermal error. If the thermal error system is linear, this can be represented as follows

$$y_t = a_1 x_{t1} + a_2 x_{t2} + \dots + a_n x_{tn} + \varepsilon_t \quad (1.a)$$

$$t = 1, 2, \dots, N$$

where, y_t is thermal error, x_{ti} is temperature in each location, a_i is the coefficient of model, ε_t is the difference between measured error and predicted error at time t .

Equation (1.a) is a typical multi-input and multi-system. where, the numerous temperature data are input, and the 5 spindle thermal errors are output. Then Equation (1.a) is modified as follows

$$Y = \begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ \vdots \\ y_N \end{bmatrix}, X = \begin{bmatrix} x_{11} & x_{12} & \dots & x_{1j} & \dots & x_{1n} \\ x_{21} & x_{22} & \dots & x_{2j} & \dots & x_{2n} \\ \vdots & \vdots & & \vdots & & \vdots \\ x_{i1} & x_{i2} & \dots & x_{ij} & \dots & x_{in} \\ \vdots & \vdots & & \vdots & \ddots & \vdots \\ x_{N1} & x_{N2} & \dots & x_{Nj} & \dots & x_{Nn} \end{bmatrix}, A = \begin{bmatrix} a_1 \\ a_2 \\ a_3 \\ \vdots \\ a_n \end{bmatrix} \quad (1.b)$$

$$A = (X'X)^{-1} X'Y \quad (1.c)$$

$$\sigma_\varepsilon^2 = \frac{1}{N} \sum_{t=1}^N (y_t - a_1 x_{t1} - \dots - a_n x_{tn})^2 \quad (1.d)$$

This model is appropriate in approximately linear system.

Thermal error model : Neural Network

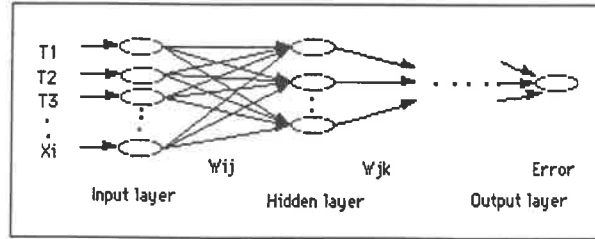


Fig.1 The component of neural network

Neural network algorithm is generally used in nonlinear system. The relation between thermal error and temperature in each location of machine tools is basically nonlinear and one set of every temperature determine the thermal error of 5 each direction. The component of neural network is input layer, hidden layer and output layer; and each layer has several nodes which have weight value and can be represented in Fig.1. The cost function(E) is defined as follows

$$E(w) = \sum_{\mu} (\xi_i^{\mu} - O_i^{\mu})^2 = \sum_{\mu} (\xi_i^{\mu} - g(\sum_k w_{ik} \xi_k^{\mu}))^2 \quad (2.a)$$

where, w_{ik} is weight value in each layer, ξ_i^{μ} is i -th measurement thermal error in μ -th measurement and O_i^{μ} is i -th predicted measurement thermal error in μ -th measurement.

In each node, summation function which is nonlinear is defined as sigmoid function as follows

$$g(x) = \frac{1}{1 + \exp(-x)} \quad (2.b)$$

When, input(temperature) and output(thermal error) are ready, the learning is processed. The learning is method of searching optimal weight value in each node. With changing the weight value, cost function(E) is calculated until minimum value of E is obtained. Amount of weight change is defined as follows

$$\Delta w_{ik} = \frac{-\eta}{1-\alpha} \frac{\partial E}{\partial w_{ik}} = \frac{\eta}{1-\alpha} \sum_{\mu} (\xi_i^{\mu} - O_i^{\mu}) \xi_k^{\mu} \quad (3)$$

where, α is momentum rate and η is learning rate.

With appropriate momentum rate and learning rate, learning is processed using steepest descent gradient method. Weight values represent the relation between temperature and thermal error and determine the performance of this model.

Thermal error model : AutoRegressive Moving Average(ARMA)

ARMA model is represented as follows

$$X_t = \phi X_{t-1} + a_t \quad (3.a)$$

$$X_t = \sum_{j=0}^{\infty} \phi^j a_{t-j} \quad (3.b)$$

$$X_t = \sum_{j=0}^{\infty} G_j a_{t-j} \quad (3.c)$$

this, X_t is output in time t , ϕ is coefficient of system and a_t is residual error

Equation(3.a) is modified to (3.b) and if we denote $G_j = \phi^j$, equation(3.b) is modified to (3.c).

The solution of differential equation (3.d) can be transformed to equation (3.e). The coefficient function φ^j in equation (3.b), which represents the essential part of the solution, is called the green function. If equation (3.e) is converted to numerical form, the coefficient of system which will represent the real system can be traced with the least square method. Whrease the simplest form of ARMA model which is represented equation (3.c) has first order derivative, general form of this model which is represented equation (3.f) has n-th order one. The higher order model can be applied for the more complex system.

$$\frac{dX(t)}{dt} + a_0 X(t) = Z(t) \quad (3.d)$$

$$X(t) = \int_0^x G(v)Z(t-v)dv \quad (3.e)$$

$$\begin{aligned} X_t - \phi_1 X_{t-1} - \phi_2 X_{t-2} - \dots - \phi_n X_{t-n} \\ = a_t - \theta_1 a_{t-1} - \theta_2 a_{t-2} - \dots - \theta_{n-1} a_{t-n+1} \end{aligned} \quad (3.f)$$

In general, the principal difference is that the regression model and neural network model are static whrease the ARMA systems are dynamic. Thermal error system has analogy with ARMA system because relation between input(temperature) and output(thermal error) is dynamic.

The measurement system and experiment

The component of thermal error measurement system has three module, the first is temperature measurement system using thermo couple and amplifier interfacing GPIB communication, the second is communication between PC and machine tools controller with RS-232, the third is thermal error measurement, with 5 gap sensor and amplifier. The measured distance is aquired converting voltage output amplifer to digital data using A/D converter which is implemented in PC. Fig.2 and Fig.3 shows gap sensors for spindle error and temperature location around spindle. Several pattern of operation mode of machine is tested such as random, sinusoidal, constant. Random operation data, which is a more general case, are obtained and thermal error prediction is simulated using the three model. The result are shown in Fig.4. Comparing the thermal error prediction using among three model, ARMA model has better representation than others. With predicted error which is output of regression model, error correction is performed using NC command which is on-line RS-232 communication. Fig.5 shows two sets of error data. one is error with correction the other is without correction. Wihout correction, thermal error grows up to about $50\mu\text{m}$, whrease with correction, error is reduced to within $\pm 5\mu\text{m}$ which is 20~30% of the uncorrected.

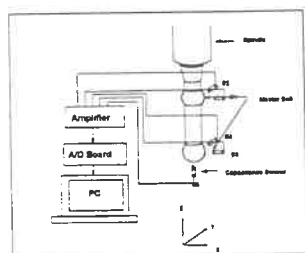
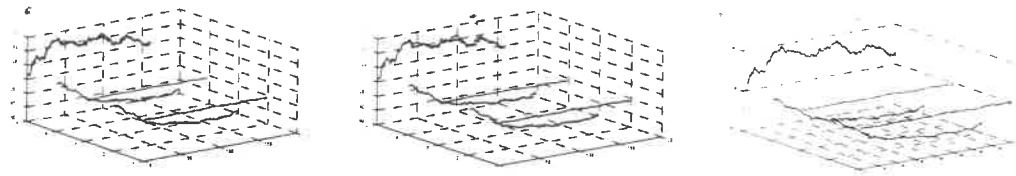


Fig.2 Sensor for spindle error



Fig.3 Temperature location



(a) Prediction by Linear regression

(b) Prediction by neural network

(c) Prediction by ARMA

Fig.4 Prediction of thermal error

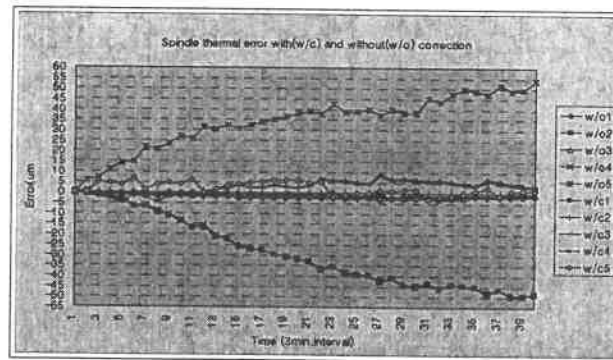


Fig.5 Thermal error with and without correction

Conclusion

In this paper, thermal error measurement/correction system has been developed using the three different model : regression model, neural network, ARMA. The correction process is developed using NC command correction.

The time optimal control of one-servo actuator with long stroke and high precision

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Abstract

A dual servo stage has been popular for the industrial purpose. However, its structure is rather complex and difficult to control. A simple structure such as linear motion table of one degree of freedom with VCM type linear actuator and air bearing suspension has been suggested. Generally one servo mechanism has a disadvantage of slow response to reach desired position. Therefore, in order to overcome this drawback, and achieve more fast response time optimal control scheme is proposed. The key concept is defining the switching surface, using time delay estimation to overcome the nonlinearity and modelling error, and then applying a proper optimal algorithm to respective switching surface. Although this system has nonlinearity or modelling error, it can bring fast and no steady state responses. With this control scheme, this system takes 0.2 sec to reach 10 mm stroke.

1. Introduction

A dual servo stage has been popular for the industrial purpose. However, its structure is rather complex and difficult to control. A simple structure having less control effort as well as nanometer resolution has been suggested[1]. The linear motion table of one

degree of freedom with VCM type linear actuator and air bearing suspension is such one. Air bearing allows the system to have no friction and wear which prohibit precision positioning due to stick slip phenomenon. Therefore, while this system moves long range, it does not require secondary fine motion control servo mechanism. Generally one servo mechanism has a disadvantage of slow response to reach desired position. In order to overcome this drawback, and achieve more fast response than that of conventional PID controller, time optimal control scheme is proposed. Bang-bang control is well known as 'best' controller[2]: Theoretically perfect control is bang-bang. This method consists of a phase of maximum acceleration toward the goal state, followed by maximum deceleration. The solution of the perfect bang-bang control strategy depends on switching time at which maximum acceleration changes to maximum deceleration. Unfortunately, the implementation of a bang-bang controller is limited by the model error. Even the small modeling error would result in the switching time error which results in bad responses. To overcome the modelling error time-delay estimation[3] is used in proposed control algorithm. The proposed control algorithm has many advantages : (1)it has robustness to modelling error which is critical to the bang-bang

control. (2) the number of tuning parameter is three (s_b, e_b, λ). (3) no chattering problem and no steady state error.

2. System Configuration

The one servo with long stroke, fine motion system consists of controller(PC), control driver, linear motor(VCM type), laser interferometer displacement sensor(Fig. 1).

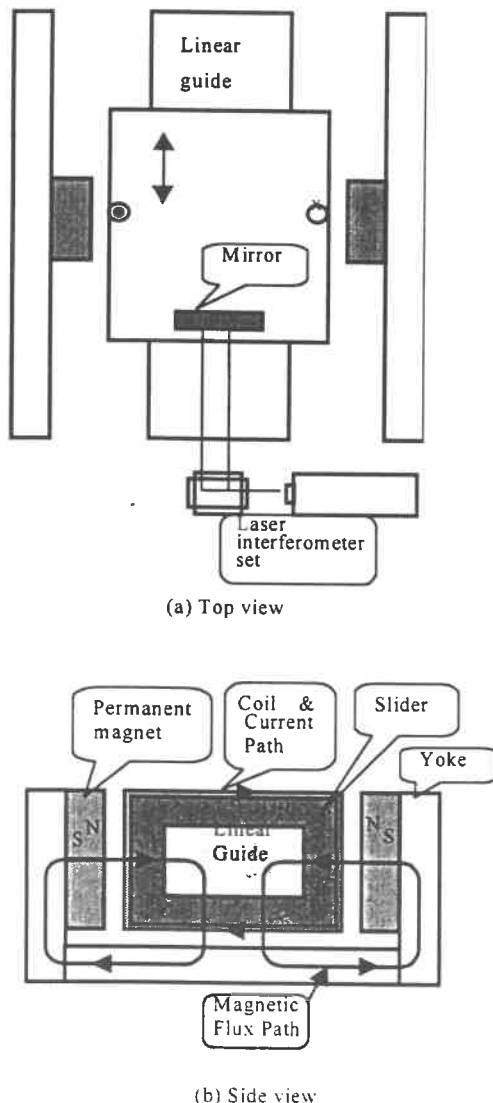


Fig. 1 System configuration

The control algorithm can be implemented at controller, and the calculated control input

drives linear motor. Linear motor is floated on the air with the aid of air bearing so that there is no friction which degrades the precision positioning capability. Displacement is measured by laser interferometer with 5nm resolution. This system can provide 100 mm stroke and 50nm resolution of motion.

3. Control strategy

3.1 Plant dynamics

On the base of parameters of the linear motor, the dynamics of the system can be described as follows:

$$L \frac{di}{dt} + Ri = V - k_b \dot{x}$$

$$M \ddot{x} = k_b i \quad (1)$$

where L is inductance, R , resistance, V , input voltage, k_b , back emf constant, M , moving mass, respectively. After taking Laplace transform and removing the current $I(s)$, the equation (1) can be :

$$\frac{x(s)}{V(s)} = \frac{k_b}{M(Ls + R)s^2 + k_b^2 s} \quad (2)$$

because R is comparatively larger than L , inductance, L , can be neglected. Then the dynamics of the system is as follows:

$$\ddot{x} + b\dot{x} = kV = u \quad (3)$$

where $k = \frac{k_b}{MR}$, $b = \frac{k_b^2}{MR}$, u is input voltage.

3.2 Optimal control algorithm

To perform a minimum-time control of the plant, a theoretically perfect control algorithm is bang-bang. This method consists of a phase of maximum acceleration toward the goal state, followed by maximum deceleration. The solution of the perfect bang-bang control strategy depends on switching time at which maximum acceleration changes to maximum

deceleration. Unfortunately, the implementation of a bang-bang controller is limited by the model error. Even small modeling error would result in the switching time error which results in bad response. To overcome the modelling error, in the proposed control algorithm, time-delay estimation is used. Based on the sliding mode control, the principle of proposed control algorithm is as follows.

Let the control input be bounded, $|u| \leq u_{sat}$.

where u_{sat} is the saturation limit of the control input as usually all the system have. Sliding surface, $s = \dot{e} + \lambda e$, is defined. From the initial point, the system is commanded to move with maximum control input until the system state reaches within the small bound of sliding surface (s_b). that is,

$$\begin{aligned} \text{for } |s| > s_b \\ u = u_{sat} * \text{sgn}(s) \end{aligned} \quad (4)$$

when the system state reaches (s_b), control algorithm is switched to the sliding mode control with time delay estimation. In sliding mode control, to preserve the sliding surface $s = 0$, control input is calculated from $\frac{ds}{dt} = 0$.

If we apply to our plant,

$$\begin{aligned} \frac{ds}{dt} &= \dot{e} + \lambda \dot{e} = \ddot{x}_d - \ddot{x} + \lambda \dot{e} \\ &= \ddot{x}_d - (u - b\dot{x}) + \lambda \dot{e} = 0 \end{aligned} \quad (5)$$

$$u = b\dot{x} + \ddot{x}_d + \lambda \dot{e} \quad (6)$$

where b is the damping coefficient of the plant which can not be estimated exactly. Therefore, using the time delay estimation, we do not have to know the exact value of b .

$$b\dot{x}(t) \approx b\dot{x}(t-dt) = u(t-dt) - \ddot{x}(t-dt) \quad (7)$$

with equation (7), equation (6) will be

$$u(t) = u(t-dt) + \ddot{x}_d + \lambda \dot{e} - \ddot{x}(t-dt) \quad (8)$$

By the way, because above algorithm has the steady state error, we put another term which make the system have no steady state error. To summarize above explanation,

$$\begin{aligned} \text{for } |s| < s_b \\ u(t) &= u(t-dt) + \ddot{x}_d + \lambda \dot{e} - \ddot{x}(t-dt) \\ \text{for } |s| < s_b \text{ and } |e| < e_b \\ u(t) &= u(t-dt) + \ddot{x}_d + \lambda \dot{e} - \ddot{x}(t-dt) + \lambda s \end{aligned} \quad (9)$$

where e_b is small error bound, and λs is for no steady state error.

The proposed control algorithm has many advantages: (1)It has robustness to modelling error which is critical to the bang-bang control. (2)The number of tuning parameter is three (s_b, e_b, λ) (3)No chattering problem and no steady state error.

3.3 Simulation result and discussion

Generally, one third of maximum stroke is most available. However, to explain the performance of the system, it is commanded to move 10mm as fast as possible. λ is adjusted according to the system. System parameters for simulation are given in table 1.

Parameters	Values
Moving mass (M)	1.708 (kg)
Coil resistance (R)	3.33 (ohms)
Coil inductance (L)	85 (mH)
Back emf constant (k_b)	1.580 (V · s/m)
Sampling time (dt)	0.0005 (sec)
Control input limit (u_{sat})	10 (V)
Switching band (s_b)	0.1
Error band (e_b)	0.01
Lambda (λ)	50

Table 1. Parameters of the system.

As shown in the Fig. 2, the time for the

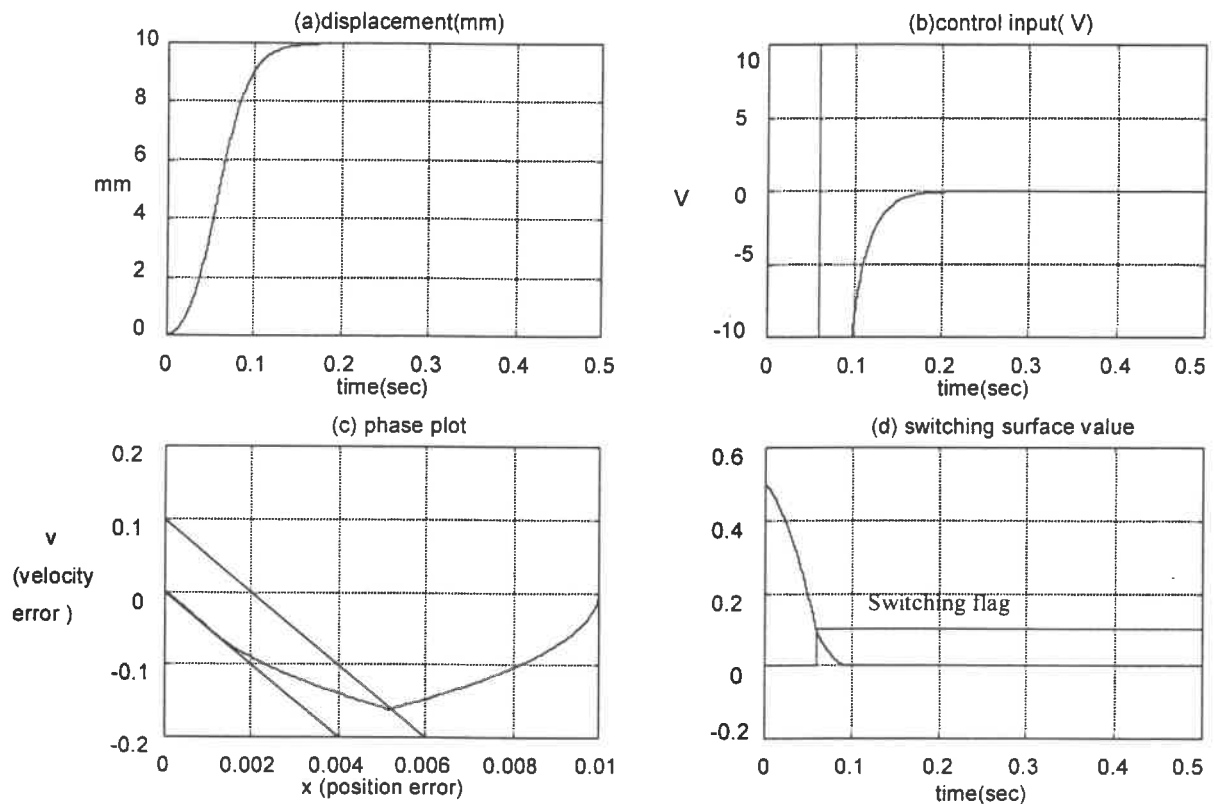


Fig. 2 Simulation results of proposed control algorithm

system to reach to within the resolution level is 0.2 sec. It is shorter than conventional PID control. In Fig. 2 (c), from initial point, as states meet the upper line, control algorithm is changed. However, because of the lower limit of control input, states go along the time optimal path(bang-bang control trajectory) until it meets the upper line in Fig 2. (c). It can be said that this system is moved along the time optimal path almost total stroke and moved along the sliding surface last short stroke to bring no chattering and no steady state error.

4. Conclusion

The new control algorithm for time optimal control of one servo actuator with long stroke and high precision is proposed. With the simulation result, we can obtain near time optimal responses such that this system could

be moved 10mm within 0.2 sec. The proposed algorithm has several advantages of robustness to the modelling error which is critical to the bang-bang control, small number of tuning parameter and no chattering problem and no steady state error.

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Machining Error Compensation of External Cylindrical Grinding Using Thermally Actuated Rest

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Keyword Thermal Error Compensation, Cylindrical Grinding Machine, Work Rest, Heat Actuator, Genetic Algorithm

1. Introduction

In grinding a long slender type workpiece, such as ballscrew, by the external cylindrical grinding machine, the static deflection of the workpiece and the thermal deformation of the machine due to environmental temperature fluctuation, internal heat sources and grinding heat cause a significant machining error [7]. To achieve the stability of machining accuracy for the prolonged operation, not only the grinding machine design and the fabrication are very important, but also compensation for errors resulted from thermal deformation of machine structure is strongly required. In order to compensate these errors in ballscrew fine grinding, an actuator to be operated within the range of 10 μ m with good reliability and smooth motion is required for getting the smooth surface. Conventionally the long slender workpiece in machining needs to be supported by one or more rests to prevent static deflection and vibration. Without deteriorating any mechanical performance of rest, the heat actuator was developed.

In this paper, at first the optimal position of the rest was investigated in order to aim at minimizing the straightness error due to the static deflection, by taking compliance of the workpiece and the structure into account. In order to obtain the optimal position of rest, a new modeling was established. Since it is so complicated to obtain the optimal position analytically for various conditions due to discontinuity, a genetic algorithm was utilized. Secondly, by applying the heat actuator upon this optimal position of the rests, the considerable effect of the thermal error compensation was achieved in the experiments.

2. Design of Heat Actuator

To develop a rest which compensate the thermal displacement and the grinding error, very sophisticatedly and smoothly operated actuator is required. A heat actuator using thermoelectric device is designed, which is a actuating bar with a flange at heating side. As the flange is cooled or heated, the actuator bar is elongated or contracted in small range of length, which will be utilized for precision positioning. The thermoelectric device uses Peltier effect to be heat or cool at the junction of 2 metals when current flows through it. By changing the current direction, the heated flange surface can be change to be cooled. As an experimental model of a practical rest, a device is designed as shown in Fig.1, of which bar material is selected by copper. In the experiment it can achieve the range of 10 μ m within a minute. From the result of the frequency response method using the swept sine mode for this device, the transfer function of the displacement output against the current input is represented as Eq.(1). The plant transfer function has integral property and time lag, also is stable having poles on the left side in s plane.

$$G_r(s) = \frac{0.0020(s+0.4357)}{(s+0.0123)(s+0.1587)} \quad (1)$$

In order to use this heat actuating system in compensating for both thermal error and deflection error in the radial direction as designated in Table 1, PI controller is used as described Eq.(2).

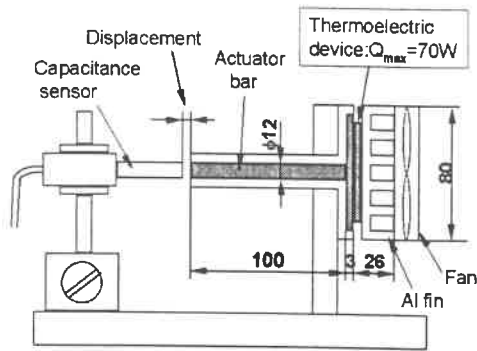


Fig. 1 Structure of heat actuator

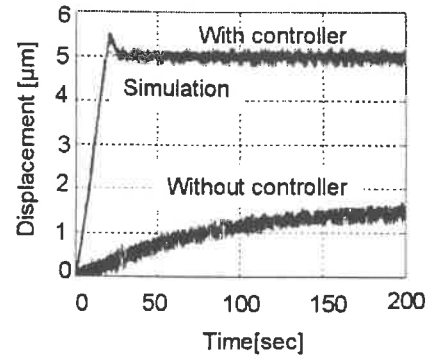


Fig. 2 5μm step response of heat actuator

$$G_c(s) = 450 + \frac{0.1668}{s}$$

(2)

Fig. 2 shows the response of 5μm step input of the heat actuating system. From the figure the result of experiment with PI controller coincides with simulation, and satisfies the specification of the system.

Table 1 Design specification and Characteristics of heat actuator with PI controller

Performance characteristics	Specification	PI
Step input, Steady state error, e_{ss} [%]	≤ 0.5	0
Maximum percentage overshoot, M_p [%]	≤ 5	1.65
Settling time, t_s [sec] (5μm step input)	60	23.5

3. Optimum Positioning of Rests

When grinding force acts upon the long slender workpiece in traverse cylindrical grinding, to reduce the workpiece shape error due to the static deflection and to control the radial displacement of rests, not only the compliance of workpiece but also that of spindle headstock, tail stock and rests should be considered [1,2,3,4]. Fig. 3 shows the modeling in which the workpiece is supported with two rests, headstock and tailstock in traverse grinding. The goal of the model is to calculate the deflection of workpiece of grinding point at x as the grinding force p moves from 0 to L . The reverse of this deflection feature becomes to the shape error.

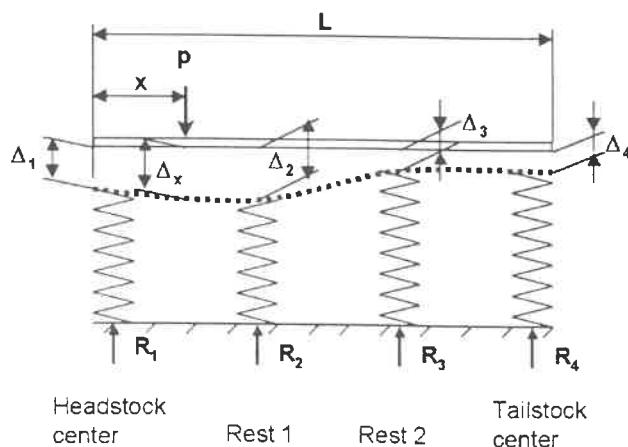


Fig. 3 Mathematical model using two rests in traverse grinding

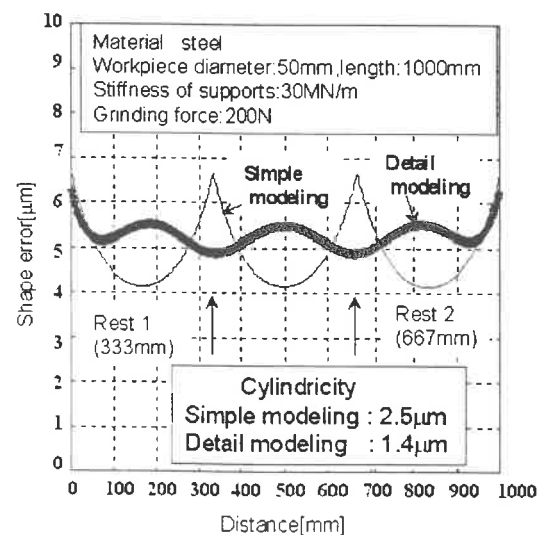


Fig. 4 Workpiece shape error with new rest positions

Since the new modeling is based on the statically indeterminate structure, each reaction forces R_2 and R_3 of the rest 1 and 2 needs to be decomposed as the redundant to act through simple beam, to apply the consistent deformations method. With this method, the deformations of rests acting the redundant R_2 or R_3 should be consistent with the deformation of workpiece by grinding force.

$$\begin{aligned} \Delta_{20} + f_{22}R_2 + f_{23}R_3 &= \Delta_2 = \frac{R_2}{K_2} & \text{where, } \Delta_{i0} : \text{deflection at } i \text{ by the force } p, \\ \Delta_{30} + f_{32}R_2 + f_{33}R_3 &= \Delta_3 = \frac{R_3}{K_3} & f_{ij} : \text{flexibility coefficient at } i \text{ by the force } R_j \\ & & K_i : \text{stiffness of rest at } i \end{aligned} \quad (3)$$

From Eq.(3), the redundant R_2 and R_3 can be determined, thus other reaction forces R_1 and R_4 are calculated by the determinate beam conditions. Defining Δ_x as a deformation at the position on which the grinding force p acts, it is calculated in Eq. (4).

$$\Delta_x = \Delta_{x0} + \Delta_{R2x} + \Delta_{R3x} \quad (4)$$

where, Δ_{x0} , Δ_{R2x} , Δ_{R3x} : deflection of the workpiece by grinding force p , reaction force R_2 , R_3

In Fig. 4, the workpiece shape error by the conventional simple model [4] and the detailed one are shown in case that all stiffness of the rests are identical. In conventional simple modeling, since the deflections of the rests are not considered and the positions of rests are discontinuous, the modeling error is predicted on those positions. On the other hand, in the new detailed modeling, since the deformations of both centers on which all grinding force are loaded are big and the deformations of rests are relatively small with the aid of other neighboring rests, the shape error is smaller, and also the overall feature is different from the simple modeling.

Next, to minimize the shape error in terms of the workpiece cylindricity, the optimum positions of two rests are determined by using the genetic algorithm [5,6]. In the genetic algorithm, the fitness is defined in Eq. (5), when the object function is radial cylindricity of workpiece, i.e. $\Delta_{\max} - \Delta_{\min}$.

$$F = C - (\max(\Delta_x) - \min(\Delta_x)) \quad (5)$$

where, C : an enough large constant

When it is repeated many times in the searching program, it reaches the optimum value in 20 generations. Table 2 represents the optimum positions for the various stiffness of rests by this method. According to the result of the simulation in the traverse grinding, it is recommended that the stiffness of both centers should be higher than those of rests.

Table 2 Optimal position and effect in the various stiffness of support by genetic algorithms

Stiffness (MN/m)			Optimum positions(mm)	Cylindricity (μm)	Improvement to equip-distance(%)
Hedstock	Tailstock	Rests			
30	30	30	317	1.33	7.0
30	30	20	346	1.41	27.5
30	30	40	297	2.24	7.1
30	20	30	375	3.51	15.0

(workpiece : steel, diameter 50mm, length 1000mm, grinding force : 200N)

4. Compensation of Thermal and Elastic Deformation

With using the test machine as shown in Fig.7, the heat actuators of rests compensate the distance between the radial position of the workpiece and the spots of rests. In every traverse stroke, the distance between the grinding wheel and the workpiece is measured by the noncontact means, which is used to

calculate the position input of rest spot. Fig. 8 shows the compensating result of the discrepancy of the wheel and the workpiece and thus the remarkable improvement of cylindricity can be noticed.

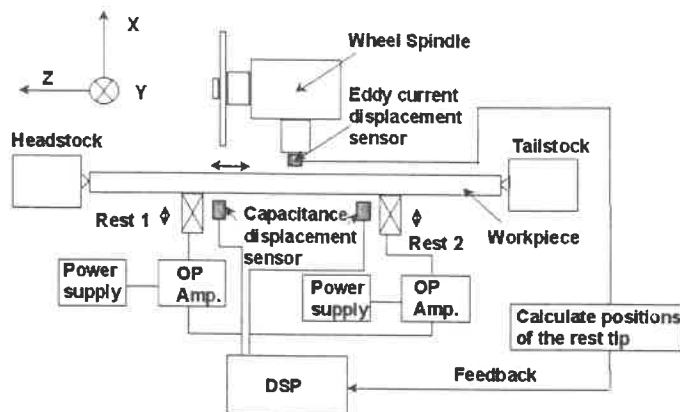


Fig. 7 Experimental setup for thermal and elastic compensation

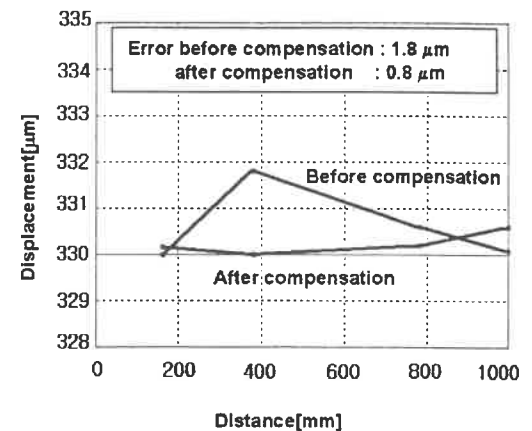


Fig. 8 Compensation of the table linear motion

5. Conclusion

In this study, to compensate thermal deformation and static deformation of a grinding machine, a control device with thermoelectric elements and thermally actuated bar which can accomplish the precise positioning having the stroke of $10\mu\text{m}$ is suggested.

In order to find the optimal position of workpiece rests, the new modeling of workpiece shape error in the external cylindrical grinding machine is established, in which the compliance of both centers and rests are considered. Based upon this new mathematical model, to minimize the cylindricity error of finished workpiece, the optimal positions of the rests are investigated with using the genetic algorithm regarding the various combinations of such compliance. As the result of the simulation about the traverse grinding, the stiffness of the rests is required to be smaller than those of both centers.

On the positions of the rests which are obtained by prescribed method, as the result of compensating experiment by the suggested heat actuator regarding the discrepancy between the grinding wheel and the rests, the cylindricity of the finished workpiece is remarkably improved.

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COMPENSATION OF NONLINEAR FRICTION FOR HIGH PRECISION POSITIONING IN THE BALL SCREW DRIVEN SLIDE SYSTEM

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Introduction

Friction is an important aspect for high quality servo mechanism. It complicates the controller design process and can significantly reduce the positioning accuracy. Control strategies that attempt to compensate for the effects of friction require a suitable friction model to predict and to compensate for the friction. Most of the existing model-based friction compensation schemes use classical friction models, such as Coulomb and viscous friction. In applications with high precision positioning and with low velocity tracking, the results are not always satisfactory. The classical models explain neither hysteretic behavior nor small displacements that occur at the contact interfaces during stiction. A better description of the friction phenomena for low velocities can be a connection of a stiff spring and damper [1].

Friction in a ball screw can reduce the capability of a drive system to position a ball-screw driven slide to sub-micrometer accuracy. This friction behaves like a variable non-linear pitch at the reversal points of the drive system making it difficult to arrive at the desired position. A dual mode dynamic friction model was used for model reference adaptive control of ball screw driven positioning system, with improved performance as the result [2].

A single stage dynamic friction model that captures the spring like motion during the stiction regime was proposed by C. Canudas, et. al [3].

PID controller incorporated with friction estimate based on the new friction model was devised and its performances were investigated through experiments.

Characterization

The experimental system setup shown in Fig. 1 consists of a PMI DC Servo Disc Motor (S9M4H) attached to the ball-screw shaft through coupling, and the motor is supplied with currents by PMI PWM DC Servo Amplifier (VXA series) according to the control voltage from DAC of the PC-31 DSP controller. A piezoelectric stack from Kinetic Ceramics, Inc. is installed between the flexure and mounting block and is preloaded to avoid fracture due to tensile stresses. The slide is supported on the linear cross roller bearing. Position signal is fed back through a capacitance sensor which provides an analog signal to the controller. The whole system is mounted on an air table for vibration isolation and are located in a temperature controlled room.

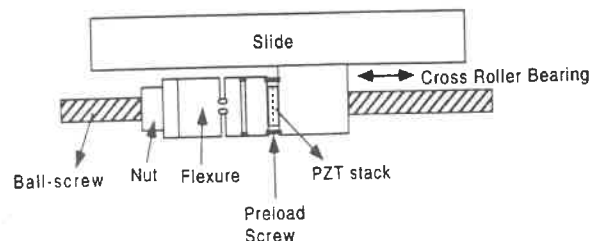


Fig. 1 : Experimental setup

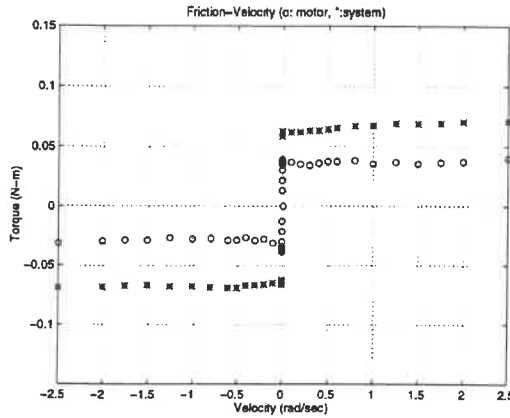


Fig. 2 : Static friction-velocity map of the motor and system

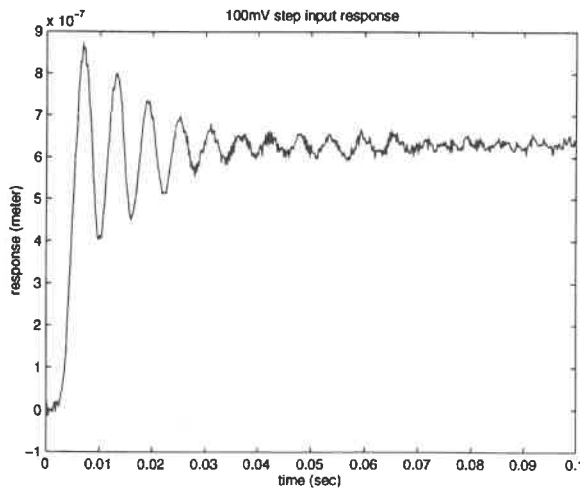


Fig. 3 : Open-loop response to 100mV step input

The static friction parameter can be estimated from the friction-velocity plot measured during constant velocity motions (Fig. 2). At a constant velocity all of the torque generated by the motor is used to cancel the friction. In this work, closed loop experiments under velocity PI control were performed. The sampling time was 200 μ s. The friction-velocity data values are then obtained by averaging the measured velocity and input torque values. This average is needed owing to velocity noise and ripple torque.

To investigate the positioning capability, the open-loop step input response was tested and the result is shown in Fig. 3.

A New Friction Model

The new model should express the classical friction behavior as well as microdynamic elastic friction of the interfaces.

A new friction model including the spring-like behavior is illustrated in Fig. 4 [3]. The interface can be modeled as two rigid bodies that make contact through elastic bristles. When a tangential force is applied, the bristles will deflect like springs which give rise to the friction force. If the force is sufficiently large some of the bristles deflect so much that they start slipping. The model proposed is based on the average behavior of the bristles. The average deflection of the bristles is denoted by z and is modeled by

$$\frac{dz}{dt} = v - \frac{|v|}{g(v)} \sigma_0 z \quad (1)$$

where v is the relative velocity between the two surfaces.

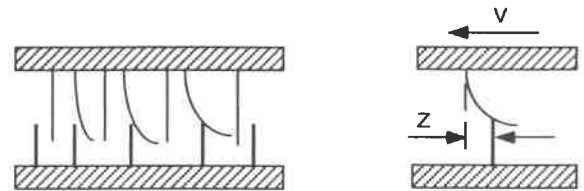


Fig. 4 : A new friction model

The friction force generated from the bending of the bristles is described as

$$F = \sigma_0 z + \sigma_1 \frac{dz}{dt} \quad (2)$$

where σ_0 is the stiffness and σ_1 is a damping coefficient.

A term proportional to the relative velocity could be added to the friction force to account for the viscous friction so that

$$F = \sigma_0 z + \sigma_1 \frac{dz}{dt} + \sigma_2 v \quad (3)$$

where σ_2 is a viscous friction coefficient.

The function $g(v)$ can be determined by measuring the steady-state friction force when the velocity is held constant,

$$F_{ss} = \sigma_0 z_{ss} + \sigma_2 v = F_C \text{sgn}(v) + \sigma_2 v \quad (4)$$

which is a usual classical model of combination of Coulomb friction, F_C , and viscous friction.

Design of Controller

Since the desired bandwidth of the system is very low compared to the natural frequency of the flexure-slide mass, the servo-system model is simplified to ball screw dynamics

$$J\ddot{\theta}(t) + F(\cdot) = u(t) \quad (5)$$

where J is a rotational moment of inertia, $\theta(t)$ is the angular displacement of the ball screw, and F is a friction torque.

As shown in Fig. 5, the control input $u(t)$ is designed as:

$$u(t) = u_{PID} + J\ddot{\theta}_r + \hat{F} \quad (6)$$

where $\ddot{\theta}_r(t) = \ddot{\theta}_d(t) + \lambda \dot{e}(t)$ and $\hat{F}$ is a friction torque estimate based on the new friction model.

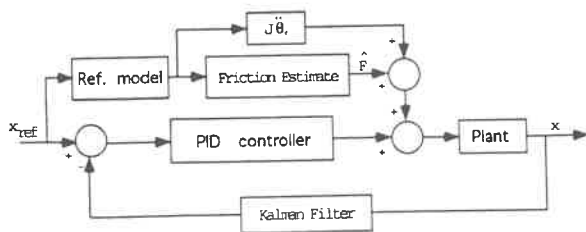


Fig. 5 : Control block diagram

Controller parameters for implementation was estimated by open-loop experiments. Coulomb friction torque, F_C and viscous friction coefficient, σ_2 are approximated using the torque-velocity relation in Fig 2. In

the stiction regime ($z \approx \theta$, $v \approx \frac{d\theta}{dt}$), the linearized description of the system (5) can be approximated by

$$J\ddot{\theta} + (\sigma_1 + \sigma_2)\dot{\theta} + \sigma_0\theta = u(t). \quad (7)$$

This equation can be used to approximate values for σ_0 and σ_1 by matching its step input response to the system position measurement (Fig. 3). Kalman filter was developed to provide good approximate of slide velocity from noisy position signal. Reference model was selected to provide a rise time of 0.1 seconds and a slightly overdamped response.

Experimental Results

The control scheme was implemented on a PC-31 (TMS320C31) DSP controller. The algorithm is written in C and installed as a interrupt service routine at 5 KHz sampling. For fairness of comparison, the same PID gains were used for all cases. Figures 6-8 display very similar responses, regardless of the magnitude of the step. The performances of the controller without friction estimate were compared to the responses of the proposed controller at each step command. Friction estimate tends to cancel the friction in the system so the resulting behavior of the system follows that of the linear system. On the other hand, controller without friction estimate can reduce the error caused by friction only by the integrating action of the controller, resulting in long time. As the step command decreases, the system response deteriorates because the position signal noise level compared to command is increasing, thus making the system sensitive to noise. A series of experiments shows that the controller performance is varying along the shaft as the ball-screw is rotating.

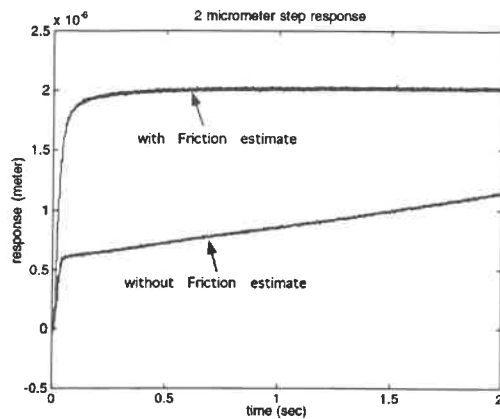


Fig. 6 : Closed-loop response to 2 μm step command

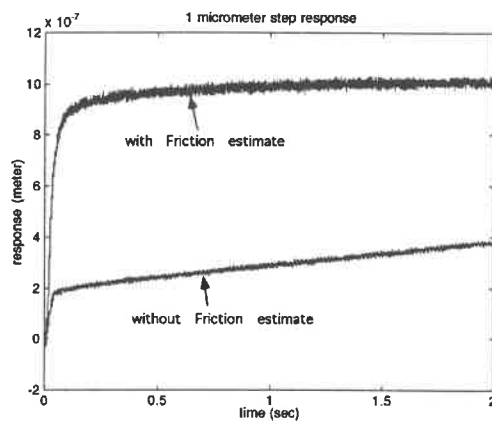


Fig.7 : Closed-loop response to 1 μm step command

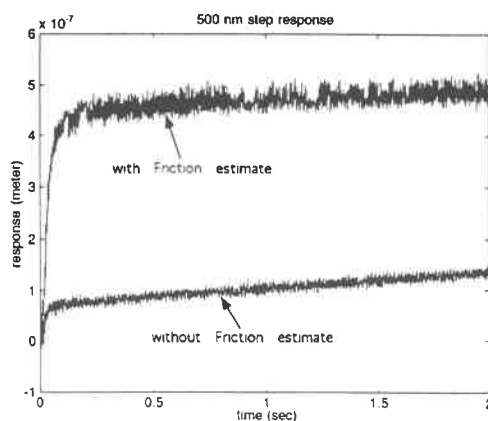


Fig.8 : Closed-loop response to 500 nm step command

Conclusion

The development and implementation of a PID controller incorporated with friction estimate based on the new friction model was presented. Experimental results verified the effectiveness of the friction estimate. Friction torque varies along the shaft so the adaptive mechanism is required of the proposed controller.

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Eigenstructure Assignment Techniques for Precision Motion Control

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Abstract

Precision motion control in devices such as photolithographic steppers, precision grinders and diamond turning machines typically involves the need for precise and rapid positioning capabilities as well as good isolation from the environment. Due to increasing requirements on the throughput of these machines, the transients due to various motions and disturbances become very important. To control the transients throughout the machine, it is important to control the closed-loop eigenvectors in addition to the closed-loop eigenvalues. Such an approach is called eigenstructure assignment. In this paper, a canonical model is created for such precision machines and a novel procedure for eigenstructure assignment is illustrated using this model.

Introduction

Typically, the control of precision machines has been performed by resorting to classical single input-single output (SISO) techniques [1]. SISO strategies for control usually do not address the coupling between the various parts of the machine and the control is hence restricted in terms of bandwidth. Good control performance requires that the dynamics of the machine be controlled as a whole rather than by closing SISO loops around various parts of the machine. This is done by application of multi input-multi output (MIMO) control strategies. MIMO control typically has been performed by the application of linear quadratic techniques and more recently using H_∞ techniques. However, these methods require that the user come up with weighting matrices. No ground rules have been established for the selection of these weighting matrices for an arbitrary control problem and there is a significant lack of intuition as the order of the problem gets large.

In this paper, we propose an alternative Eigenstructure Assignment technique where the desired modal properties of the closed loop system are specified. The resulting inverse eigenvalue problem is solved. Eigenstructure assignment is especially applicable to precision machines since a great deal of effort is spent in designing the machine to have good dynamical properties and because very good models can be obtained for the machine. Thus model-based controllers can be designed to have high bandwidths instead of restricting ourselves to low bandwidth SISO loops. It is well known that the closed-loop eigenvalues of the machine determine its speed of response and the stability, while the closed-loop eigenvectors determine the transient responses. To have good control over the machine and its transients, it is necessary to obtain the desired eigenvalues (natural frequencies) and eigenvectors (mode shapes) for the machine. However, this might not always be possible. We discuss the assignment problem and its solution in the subsequent sections.

Problem Statement

Consider the state space representation of a system given by

$$\begin{aligned}\dot{\mathbf{x}} &= \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u} \\ \mathbf{y} &= \mathbf{C}\mathbf{x} + \mathbf{D}\mathbf{u}\end{aligned}\tag{1}$$

where $\mathbf{A}$, $\mathbf{B}$, $\mathbf{C}$ and $\mathbf{D}$ are the system matrices, $\mathbf{x} \in \mathbb{R}^n$ the state vector, $\mathbf{u} \in \mathbb{R}^m$ the control input vector and $\mathbf{y} \in \mathbb{R}^p$ the output vector. We consider the state feedback regulator problem. The feedback control law is given by

$$\mathbf{u} = -\mathbf{K}\mathbf{x}\tag{2}$$

where $\mathbf{K} \in \mathbb{R}^{m \times n}$ is the control gain matrix that is to be determined. Substituting the control law given by (2) into the state equations given in (1) leads to the following closed loop state equations

$$\begin{aligned}\dot{\mathbf{x}} &= \mathbf{A}_c\mathbf{x} \\ \text{where} \\ \mathbf{A}_c &= \mathbf{A} - \mathbf{B}\mathbf{K}\end{aligned}\tag{3}$$

The problem of eigenstructure assignment is to assign as closely as possible the desired eigenvalues λ_i^d and eigenvectors $\mathbf{v}_i^d$ to the closed loop system above. We first express the attainable eigenvalues and eigenvectors in terms of the desired ones as follows :

$$\lambda_i = \lambda_i^d + l_i \quad (4)$$

$$\mathbf{v}_i = \mathbf{v}_i^d + \mathbf{e}_i \quad (5)$$

l_i and $\mathbf{e}_i$ are the errors in between the attainable and desired eigenvalues and eigenvectors respectively, and our task is to minimize these errors. Given that we have a self-conjugate set of eigenvalues and eigenvectors, we have the familiar eigenvalue problem

$$\begin{aligned} (\lambda_i \mathbf{I} - \mathbf{A}_c) \mathbf{v}_i &= 0 \\ \Rightarrow (\lambda_i \mathbf{I} - \mathbf{A} + \mathbf{BK}) \mathbf{v}_i &= 0 \end{aligned} \quad (6)$$

Now, let $\mathbf{K}\mathbf{v}_i = \mathbf{u}_i$ such that movements $\mathbf{u}_i$ in the control space cause movements $\mathbf{v}_i$ in the state space. Substituting this expression into the (6) then leads to

$$\mathbf{L}_i \mathbf{u}_i = \mathbf{v}_i \quad (7)$$

where

$$\mathbf{L}_i = -(\lambda_i \mathbf{I} - \mathbf{A})^{-1} \mathbf{B} \quad (8)$$

and we have assumed that the closed-loop eigenvalues are different from the open-loop eigenvalues. Also, by our earlier definition, the control gain is given by

$$\mathbf{K} = [\mathbf{u}_1, \mathbf{u}_2, \dots, \mathbf{u}_n] [\mathbf{v}_1, \mathbf{v}_2, \dots, \mathbf{v}_n]^{-1} \quad (9)$$

From the above discussion, we note that the eigenvalues and the eigenvectors cannot be specified arbitrarily. They need to satisfy the constraint equation (7). We thus have n equations in m unknowns, with the number of states n typically much greater than the number of control inputs m leading to an overconstrained problem. This equation is usually inconsistent for arbitrary choices of λ_i^d and $\mathbf{v}_i^d$. Typically, eigenstructure assignment has been carried out by placing the eigenvalues of the closed loop system at some desired locations and obtaining the best fit eigenvectors (usually according to a least squares error criterion). Various authors have dealt with this problem and it is well documented in the literature [2]. This has largely been applied to aerospace problems and we now show the application of this technique to precision motion control applications.

Eigenvalue Assignment

We first note a few facts about the constraint equations before proceeding with the solution of the problem. We see that to obtain the desired closed loop eigenvectors $\mathbf{v}_i^d$ exactly, they must reside in the subspace spanned by the columns of $\mathbf{L}_i$. This subspace has dimension m (which is the rank of $\mathbf{B}$ i.e., the number of independent control variables. We observe the following :

- The number of independent control variables determines the dimension of the subspace in which the achievable eigenvectors must reside
- The orientation of this subspace is determined by open-loop system matrices $\mathbf{A}$ and $\mathbf{B}$ and by the desired closed-loop eigenvalues λ_i .

In general, a desired eigenvector $\mathbf{v}_i^d$ may not reside in its prescribed subspace and hence we look for a best fit to this eigenvector. This best fit in the sense of least squares (minimum Euclidean norm of the error $\mathbf{e}_i$) is given by projecting $\mathbf{v}_i^d$ onto the subspace spanned by the columns of $\mathbf{L}_i$. This leads to the normal equations given by

$$\begin{aligned} \mathbf{u}_i &= (\mathbf{L}_i' \mathbf{L}_i)^{-1} \mathbf{L}_i' \mathbf{v}_i^d \\ \mathbf{v}_i &= \mathbf{L}_i \mathbf{u}_i \end{aligned} \quad (10)$$

The vectors $\mathbf{u}_i$ and $\mathbf{v}_i$ are then assembled to obtain the control gain using the expression in (9).

We realize that we have only m degrees of freedom in specifying the eigenvector - this means that only m components of the eigenvector can be independently specified. This is usually not a significantly limiting

condition since in most large-scale systems, one is interested in specifying how only certain states behave in a given mode.

We handle this by using the so-called permutation or weighting matrices as follows. For any given mode, let r denote the number of terms in an eigenvector that are desired to be specified. When $r \leq m$, we can specify all r arbitrary terms in the eigenvector and the eigenvalue corresponding to the desired mode also by defining weighting matrices W_i for each mode. It is assumed without loss of generality that the weighting matrices have full rank r . We have

$$\begin{aligned} L_i u_i &= v_i = v_i^d + e_i \\ \Rightarrow W_i L_i u_i &= W_i v_i^d \end{aligned} \quad (11)$$

where we have used the fact that W_i has full rank to remove the error term e_i since the above system of equations is underconstrained at most having r equations in m unknowns. We can choose the least-squared solution (amongst many possible) to get

$$u_i = (L_i' W_i' W_i L_i)^{-1} L_i' W_i' W_i v_i^d \quad (12)$$

$$v_i = L_i u_i \quad (13)$$

The vectors u_i and v_i are then assembled as before to obtain the control gain as given in (9).

Precision Machine Model

We solve the control problem for a representative system of three masses, springs and dampers. This representative system is meant to simulate the motion in one axis of a precision machine such as a wafer stepper, diamond turning machine, or a coordinate measuring machine with a base, a stage and an uncontrolled resonance. It is required to isolate the machine from seismic disturbances (active soft mount) and have a high tool/stage bandwidth. Mass M_1 models the machine base. The tool/stage modeled by the mass M_2 needs to be positioned accurately with some other feature on the machine (*e.g.* spindle/lens); this other feature is modeled by mass M_3 . A schematic of the system is shown in Figure 1. The performance yardstick is

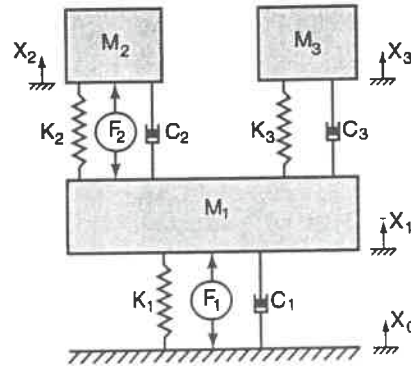


Figure 1: Canonical Model of a Precision Machine

the relative position in between the masses M_2 and M_3 . With this in mind, we identify the following three desired modes of motion :

- All three masses move in unison as a rigid body
- Masses M_2 and M_3 move together to keep the relative distance (tool-spindle distance or optical length) constant
- Mass M_2 moves freely to position itself with respect to Mass M_3

We choose the absolute positions and velocities as the state variables and convert the above description of the desired modes into a specification of eigenvectors v_i^d and weighting matrices W_i .

A numerical example has been worked out for the Z-axis of the prototype stepper described in [1]. The eigenvalues have been chosen to get a lower isolation bandwidth of 0.1 Hz, and stage motion bandwidth of

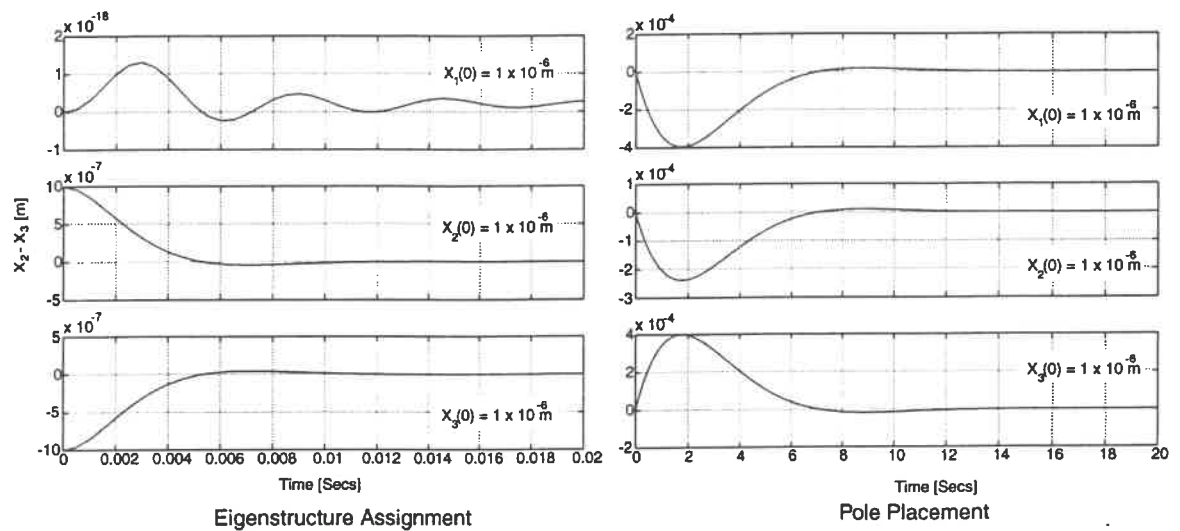


Figure 2: Comparison of Transient responses of the Closed-loop system under Eigenstructure Assignment and Pole-Placement. Note the vastly different time scales.

200 Hz. Both these modes have been chosen to be critically damped while the lens column resonance has been left untouched to avoid huge control gains. A pole placement approach using the technique described in [3] is compared with the eigenstructure assignment technique developed in this paper. The comparison is made by simulating the response of the closed loop system to initial conditions of a $1 \mu\text{m}$ deviation from equilibrium of each of the masses separately. It is seen that system designed by Eigenstructure Assignment settles within 20 ms while the system designed by Pole Placement settles within 20 s. This is in spite of both systems having the same closed-loop eigenvalues. It is thus seen that attention to the transients (or design of the closed-loop eigenvectors) is important to design fast and accurate precision machines.

Conclusion

We have shown the importance of considering both the eigenvectors and the eigenvalues to achieve fast responses in a precision machine using a canonical model for a class of precision machines. This technique is especially applicable to precision machines for which good models exist and careful attention has been paid to obtain the desired modal properties of the machine structure. There are however problems in which it is desired to specify more than m components of the eigenvector. We plan to discuss the solution to these problems in a future paper. Optimum eigenstructure assignment for the entire machine will involve eigenvalue assignment as discussed in this paper along with eigenvector assignment.

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Fundamental Ideas in Measurement Uncertainty

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The basic foundations of engineering science are the familiar concepts of mathematics (arithmetic, Euclidean geometry, analysis, etc.) and physics (conservation laws, Maxwell's equations, Newtonian mechanics, etc.). We learn these concepts in our training and they are the tools by which we communicate, logically and unambiguously, and build on the work of others. By contrast, our experience is that many engineers view probability and statistics as a hazy collection of disconnected ideas in which data analysis, beyond computing the mean and the variance of a set of repeated measurements, is the arcane domain of the professional statistician, a domain in which an answer may depend upon the method of analysis. It is natural to wonder whether there are fundamental notions here on a conceptual level equivalent to, say, Euclid's five axioms of plane geometry.

We will present a necessarily brief introduction and overview of a general and unique system of inference, or reasoning in the region between certainty and impossibility. The fundamental concept is that of probability, viewed as a real number that measures a degree of reasonable belief, and whose value is conditioned on all relevant information. Probabilities are manipulated using two simple rules that follow from elementary requirements for logical consistency and agreement with common sense. These rules show, uniquely, how a probability changes as new information becomes available, which is just the process of learning. Viewed in this light, a measurement provides information that refines and sharpens a broad probability distribution and forms the basis for a rational decision.

The algebra of inference provides no guidance for the initial assignment of probabilities. We introduce the entropy of a distribution, and describe the principle of maximum entropy which is a general means by which partial knowledge is translated into a prior probability distribution.

We will discuss engineering metrology as a special case of this general system of reasoning, and its relation to the ISO "Guide to the Expression of Uncertainty in Measurement". In particular, maximum entropy distributions for Type B standard uncertainties satisfy the natural common sense requirement that two persons with the same information should assign the same probabilities.

Application of Laser Feedback Metrology to a Hexapod Test Strut

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1. Introduction

A laser position feedback system has been designed to improve the accuracy of strut positioning for the prototype Ingersoll² Octahedral Hexapod machine installed at National Institute of Standards and Technology (NIST) (Figure 1). Strut laser feedback metrology experiments are conducted on a test strut, which duplicates one of the Hexapod's struts but is positioned in a test stand. The Octahedral Hexapod machine is based on a Stewart platform [Ref. 1], with 6 ballscrew-driven struts that support and move a cutting spindle. The position and orientation of the spindle are determined by the lengths of the struts, which extend from 2150 mm to 3600 mm. For Hexapod machines; accurate, strut-length metrology is essential to achieving the accurate spindle platform motions necessary to take advantage of the Hexapod's speed and stiffness for various machining applications. Currently, a rotational resolver on the back of each direct drive motor is used for position feedback to the machine's controller. This configuration has significant disadvantages in measuring strut length due partly to leadscrew errors, backlash errors, friction in the spherical joints, and especially thermal errors. Therefore, there is "a need for closed-loop thermally invariant metrology that addresses the length of the total strut". [Ref. 2] Direct measurement of the strut length using laser interferometry seems a promising approach to significantly increase the accuracy of the machine. The goal of this project is to develop a laser strut length metrology system that is accurate to within 5 μm . In the Conclusions, we discuss other metrology techniques under investigation.

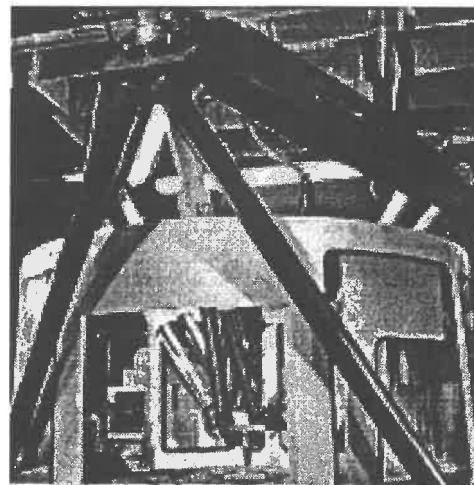


Figure 1. Hexapod Machine at NIST

2. Experiments

Experimental Apparatus

Test Strut All the tests were done on the single independent strut. Figure 2 shows a schematic of the test strut in the horizontal test position with the Pentaprism Retroreflector (PR) positioned vertically.

Lasers Two HeNe laser systems were used in experiments with the test strut. One was used to measure the distance between the large and small ball joints of the strut, and is referred to as the strut laser. It uses a single beam path, returning the measurement beam along the same path as the source. With this system there is also no separate reference beam outside the laser head. The interferometer is integrated into the laser packaging and is very close to the laser source making attachment and alignment of the optics much easier than a typical laser system [Ref. 3]. The second laser served as reference, measuring the displacement of the test stand's carriage. It has a separate interferometer package, but its operation is similar to the strut laser. The A quad B output signal from the strut laser is given to the reference laser's software as an encoder signal for comparison to the reference laser. Critical to

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achieving optimum system accuracy and repeatability [Ref. 4], both systems are compensated for environmental conditions using a weather station, which is part of the reference laser's system. Environmental precautions will be required when implementing this system on the Hexapod including protecting the laser path with telescoping tubes.

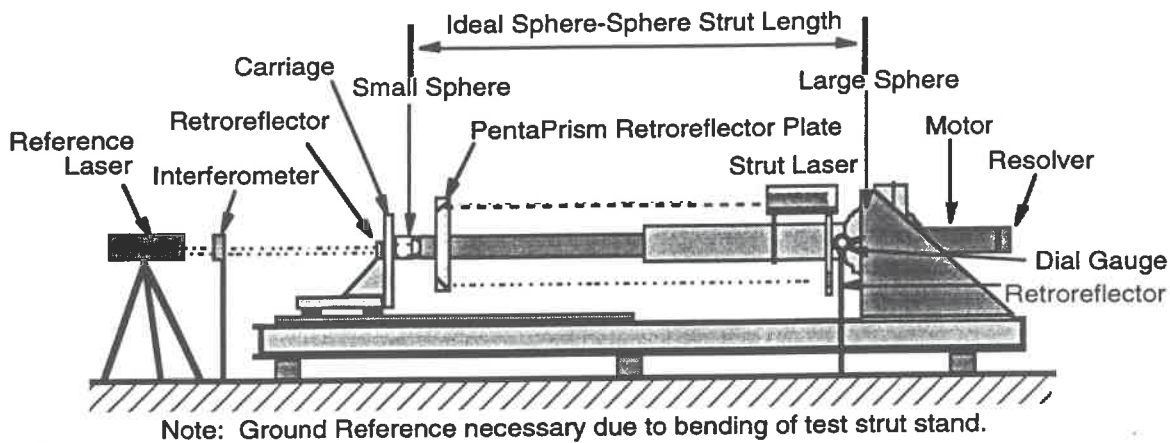


Figure 2. Schematic of Test Strut with experimental Pentaprism Retroreflector before application of a commercial Lateral Transfer Hollow Retroreflector (LTHR) and redesigned laser mounts.

Dial Gauge Two displacement transducers, as shown in Figure 3, were used to measure the displacement of the large ball joint bracket due to the sagging of the test strut stand as the strut moved throughout its range. The sagging and distortion of the test-strut-base-frame is the reason why the foundation is part of the metrology loop for the reference laser. Especially for (thermal) drift experiments, a different solution will be adopted.

Mounting Hardware A simple clamp was designed to hold an adjustable mount for the strut laser. The clamp was attached to the strut near the large ball joint. The retroreflector, for the measurement of the strut length, was located at the bottom of the clamp to return the signal from the “wrapped” laser beam. This system was later redesigned for easier laser adjustments with kinematic mounting supports and features for attaching telescoping tubes to protect the laser beam. Additional coupling brackets for the smaller tube of the test strut were designed to accept the commercial LTHR as shown in Figure 4.

Lateral Transfer Hollow Retroreflector (LTHR) A commercial Lateral Transfer Hollow Retroreflector (LTHR) shown in Figure 4 was purchased to replace the prototype pentaprism retroreflector described in the following sections and shown in Figure 5. The LTHR system functions similar to a very large thermally stable corner cube and allowed us to “wrap” the beam around and through the strut [Ref. 5].

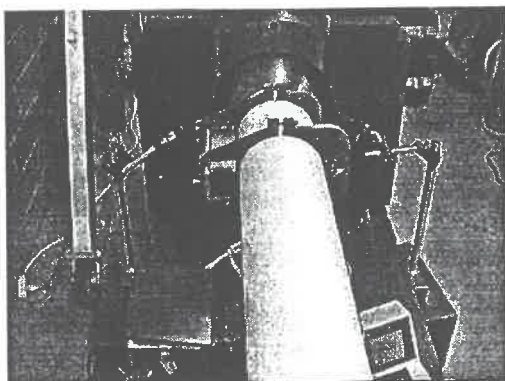


Figure 3. Test Strut showing Dial Gauges, Strut Laser and Laser Mounting bracket.

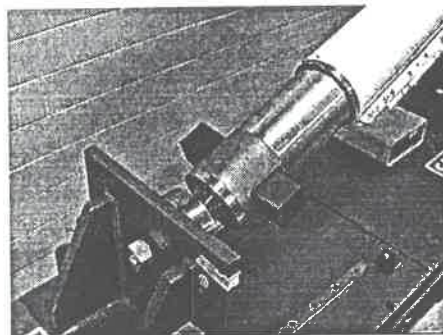


Figure 4. Lateral Transfer Hollow Retroreflector (LTHR) installed horizontally on the Hexapod Test Strut.

Background The first approach for applying laser position feedback to the strut involved a beam path through the center of the test strut. A 10 mm diameter hole was made through the center of the ballscrew and motor, with the laser head attached to the motor end of the strut. (Eventually the laser was to be mounted on a low expansion structure connected to the large sphere, thus taking the motor out of the structural loop.) The retroreflector was mounted on-axis at the small ball joint end. The beam's signal was uninterrupted for all lengths of the strut when the strut was stationary. However, under high velocity motions, the beam's signal was lost. Commercial hexapods now employ a similar technique with a larger diameter hole, which could not be done easily on NIST's Hexapod because it would require replacing the struts. One such application is Patent #5604593 by McMurty [Ref. 6].

Initial Experiments The first step was to verify that the Strut laser could be used for reliable measurements on the test strut. Many sources of error were found that caused different displacements to be measured by the strut laser and the reference laser. As they were discovered, they were eliminated if possible, or measured by other means and included in the comparison of displacement values. These preliminary results lead us to believe that it will be possible to achieve a target value of 5 μm for the uncorrected values between laser measurements. One important conclusion from these experiments was that compensation for Abbe errors [Ref. 7] due to the bending of the test strut was critical to good measurements of the strut length. Then, the solution is to take measurements on both sides and take average of the measurements.

Experiments with Wrap-around Laser Setups to Reduce Abbe Errors

Once the experiments with the strut laser showed good results, a prototype system was designed to reduce the Abbe error and to allow the measuring of both paths concurrently. A slotted steel plate (Figure 5) was designed to route the measurement beam through the strut. The design uses a pentaprism to reflect the incoming beam 90 degrees to send it through the strut and then to another pentaprism to bring it parallel to the incoming beam headed back to the large ball joint end of the strut. The bottom half of one of the clamps supporting the strut laser was modified for mounting the strut laser's retroreflector. Alignment was difficult. This is because the pentaprisms create a perfect right angle deflection in one plane, but, being 2-dimensional prisms, act as plane mirrors in another plane. It was possible to maintain sufficient signal strength over the full range of position and acceleration of the test strut. In order to assess the validity of using this laser path and mounting scheme to accurately measure strut length, measurements were performed. The reference laser was used to measure the position of the test strut carriage, while a displacement transducer was used to measure the displacement of the large ball joint bracket towards the strut. The contribution of the bracket's movement was subtracted from that observed by the reference laser, and the result was compared to half the distance measured by the strut laser. This same approach was used in a small number of preliminary experiments covering a variety of speeds and roll angles of the strut. The speeds, horizontal strut position and the roll angles are representative of the worst case scenario on the measurement system regarding the different strut orientations encountered on the hexapod. The prototype optics caused the alignment to be very sensitive, but the prototype system proved out the concept and the reason to proceed with a commercial as shown in Figure 4.

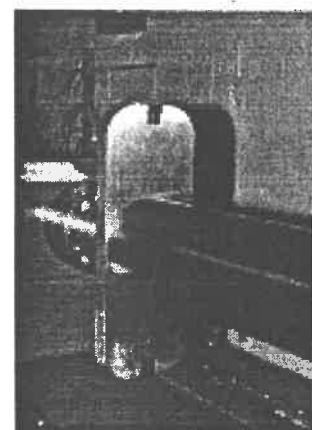


Figure 5. Pentaprism Plate

3. Conclusions

Summary Previous attempts to apply laser strut length measurement through the center of the test strut installed at NIST were not successful under dynamic conditions. An externally mounted system has been described that minimizes Abbe error by measuring along parallel paths on opposite sides of the strut. With this arrangement, bending and cocking of the extensible strut tubes have a minimal effect on the total laser path length. Tests using the laser with the pentaprism retroreflector prototype instead of the rotational resolver for position feedback confirm that the method designed decreases positioning error to around 5 μm , even as the strut, ballscrew and other components undergo considerable thermal expansion due to friction heating. In a test performed after repeated rapid motion, the steady state position error is reduced from 50 μm with the rotational resolver to less than 5 μm with the laser, over a strut travel of 1200 mm. The laser signal is maintained for all speeds and accelerations of the test strut. The prototype optics were replaced with a commercially available Lateral Transfer Hollow Retroreflector (LTHR), and new mounting clamps to bring the laser beam as close as possible to the strut centerline. Compared with the prototype pentaprism-based optics, the LTHR reduces the sensitivity to bending and cocking of the strut in the

transverse plane and facilitates alignment. Telescoping tubing will be used to protect the laser beam and optics from the flying chips and cutting fluid on the Hexapod. A sample of the tests performed with the LTHR system is shown in Figure 6. The LTHR system performed well in the horizontal position with results under 4 μm , but in

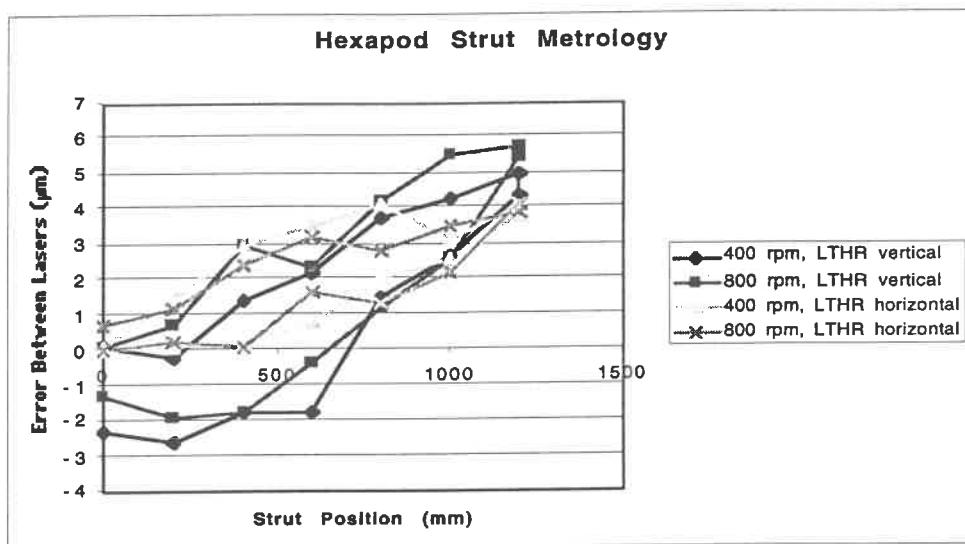


Figure 6. Preliminary and Unconfirmed Errors between the strut laser and a reference laser over the 1200 mm range for strut speeds of 8000 and 16000 mm-per-min (20 mm per revolution).

the vertical position the range of errors was up to 10 μm . These preliminary sets of data will be replaced by thorough testing following an accepted standard such as ANSI/ASME B5.54 [Ref. 8]. This initial data set may indicate a roll problem for the measurement system due to poor alignment of the retroreflector plate and laser bracket. The mounting hardware for the laser will need to be improved before full implementation to the Hexapod.

Alternative Metrology Systems Alternative metrology systems to determine the strut lengths to within 5 μm are being explored. One design is based on an internal, on-axis interferometer system by using fiber optics. Another system under consideration are linear optical encoders. One or two encoders will be attached to the external strut. This system would be very compact and eliminate the need for telescoping tubes. Laser interferometric systems will be compared to linear encoders on our test strut, while also considering other published studies conducted. [Ref. 9]

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Symbiotic System of Micro Robot and Coordinate Measuring Machine for Flexible Surface Measurement

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1. Introduction

Coordinate measuring machines and surface roughness instruments play important role because they can identify the object feature with a set of numerical data. For these years, those instruments have been much improved due to the advancement of mechanisms, materials and computation technologies. It is, however, still difficult to measure precisely the form of big artifact with the microscopic surface topography since significant mechanical stability is required. Thus it is obvious that we have to enhance the capabilities of those machines with techniques such as use of low thermal expansion material, disturbance isolation control and actuator servo manner while they need much of cost.

In our group, many micro robots with piezo elements and electromagnet for locomotion and unique manipulators have been developed, and accurate versatility such as micro-grating,

micro-indentation as well micro-sampling have been offered^{1),2),3)}. This unique efforts might be applied possibly to an automatic flexible micropart machining and assembling on the desktop⁴⁾. However there remained a problem that small robots can never jump over the obstacle while they can easily move with sub-micron step resolution using piezoelements as a source of locomotion.

It is well-known that most of animals and insects compensate each other to share the biological benefit. For instance, some insects which are parasitic on an animal can get tiny worms on its back which it can no longer shake off as well as they can be transported anywhere without being captured. (Such strategy is not so complicated but ingenious)

With adapting this symbiotic manner to the precision machinery, the combination of a coordinate measuring machine and a micro robot with surface stylus can be proposed in order to compensate the lack of performances such as the resolution of CMM and the working range of small robot. The conceptual picture is shown in Fig.1 In this report, the unique surface flexible measuring system which is ongoing development is described and a part of interesting performance is demonstrated.

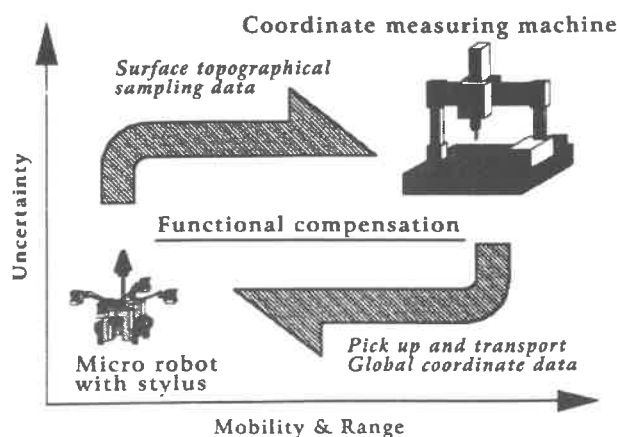


Fig.1 Functional compensation between CMM and micro surface measuring robot

2. Piezo-driven micro robot with stylus and magnetic feeler for hanging

In the previous report, the small robot with surface stylus and the camera vision coordinate monitoring system were presented for flexible surface measurement⁵⁾. With the similar locomotion mechanism, a new micro robot have been developed as illustrated in Fig.2. In order to hang on the CMM arm by itself and

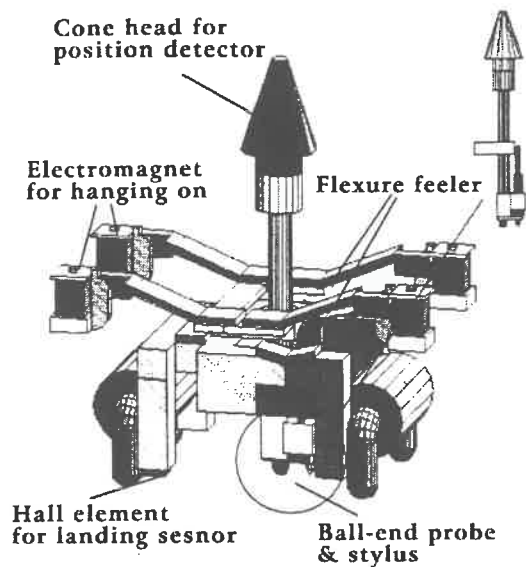


Fig.2 Piezo driven micro robot with a surface stylus, a normal probe with cone head and magnetic feelers for hanging on CMM

to measure the coordinate position of the target point, tiny electromagnets which are set on the end of each feeler from the robot body and a normal touch probe with the cone shape head and the surface stylus are equipped. The photograph is also shown in Fig.3. When the small robot is move on the target surface, it can get surface topographical data by using the stylus. Simultaneously the coordinate position of the small robot can be also monitored by the optical local position detector as mentioned in the next section. This arrangement allows mechanical isolation between two bodies but flexible connection is established on demand. The CMM can obtain both global coordinate position by reading its axis scales and local topographical gauging from the small robot read-out.

When the small robot has the task to jump into another workpiece, it can call for the CMM to come near to it. The landing deck which is set on the CMM arm can be overlaid on the small robot and in contact it, then the current to electromagnet is turned on and the robot can hang on the CMM arm automatically to be transported to the next working area.

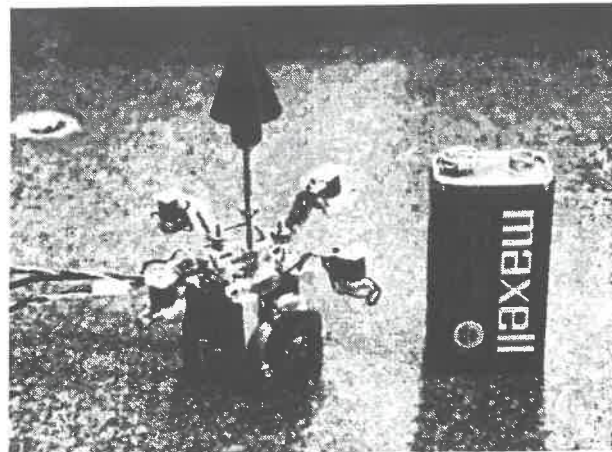


Fig.3 Photograph of the unique small robot with a cone antler and hanging feelers.

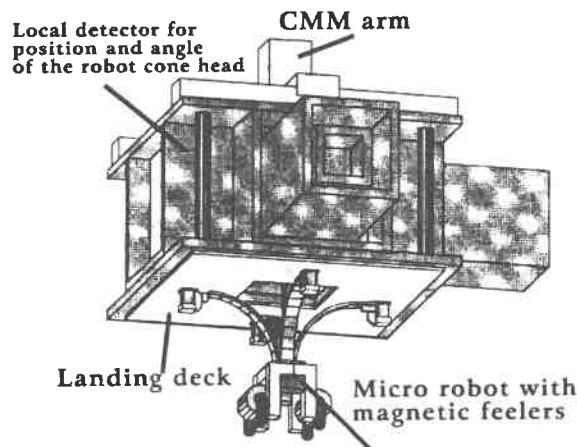


Fig.4 Local optical position detector to measure the cone target of robot and landing deck.

3. CMM and landing deck with optical local position detector

It is very important and difficult to keep the continuous accuracy over both global measuring volume and local one. On the arm of CMM, the optical position detector with a landing deck is attached as shown in Fig.4. The bottom plate of the detector enclosure has a hole so that the cone head of the robot can go through. In the detector, there are pairs of laser beam optics and CCD line image sensors which are assembled double in X-Y plane as

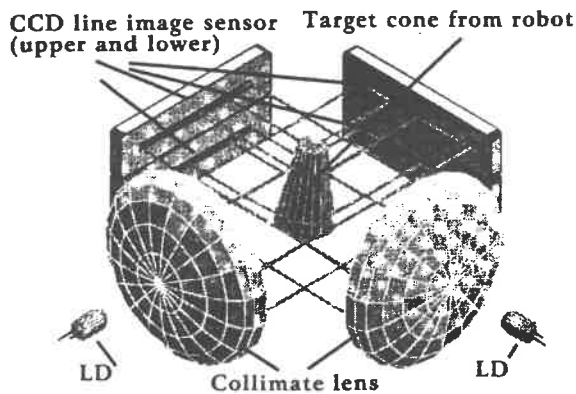


Fig.5 Optical configuration of local position detector

illustrated in Fig.5. Since the collimated laser beam are interrupted by the cone and then coordinate positions of 8 points of the edge can be obtained. These data are to be processed to get the coordinate position (x_p, y_p, z_p) of the vertex and the spatial inclination of the cone axis (θ, ϕ) . Thus the global coordinate position of the contact point of the probe on the robot is estimated by the combination of the CMM scales and sub-coordinate readings in the local position detector with the compensation of its probe angles (θ, ϕ) as shown in Fig. 6. This combination can provide good enhancement of precise performance to both CMM and micro robot instrument with low cost although much more consideration such as error analysis should be cared in order to guarantee the overall accuracy.

4. System Control and Experiment

The development of symbiotic system for micro-macro measuring task is still in the beginning stage, a part of automatic control facilities including computers and I/O interfaces have been implemented. And the appropriate sequential task-driven program can control both two machines and data acquisition. A primary experiment that the micro robot is captured and transported to another workpiece have been conducted. And there the small robot is released to move on the sample and to pick up the surface topography as shown in Fig.7 (Appropriate destination data are given by the operator because the current facility does not

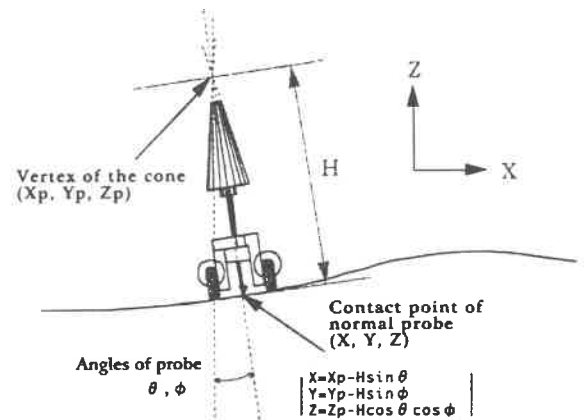


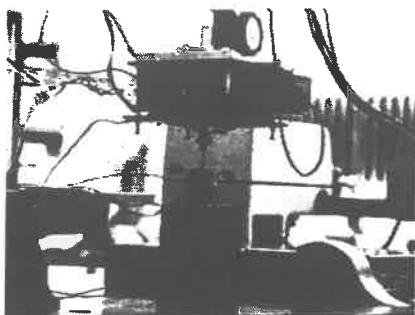
Fig.6 Vertex of the cone and contact point of the normal probe on the small robot

have the camera vision monitoring system)

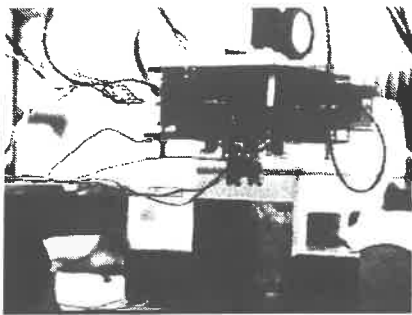
Also the drive motor of each axis of CMM is controlled to track and to keep the small robot at the center with the feedback signal from the local coordinate detector. During this task, both global coordinate position and local topographical sampling can be obtained. After it finishes to sample, then the small robot can call for the CMM and hang on it again to be transported to the initial stage. When the small robot is landing on the surface, another magnetic sensor which is attached on its leg can give a special signal of the magnetic flux. This signal is processed by the computer of CMM both to prevent the small robot colliding the surface and to exploring the landing condition. Finally the small robot can be successfully released on the initial stage.

5. Conclusion and future works

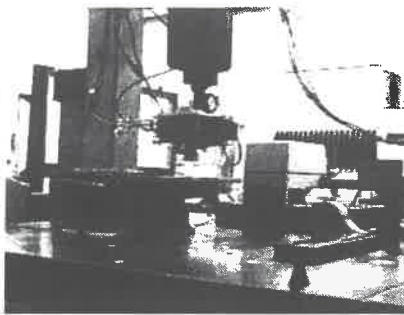
In this report, a new style of surface measuring system has been proposed for the functional compensation of a CMM and a micro robot. Here CMM can help the micro robot to expand its working range while the robot can give CMM the surface topographical data. This unique combination can provide the possibility of expanding the performance of precision machines for surface measurement with low cost and much of flexibility. Currently the overall system is being developed to establish the automatic symbiotic control for these instruments.



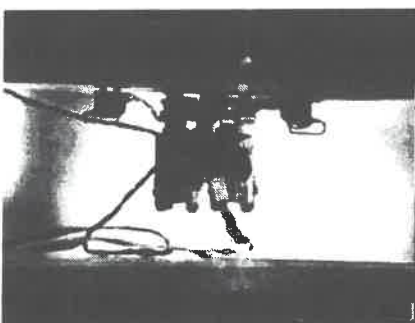
(1) Landing deck of CMM is approaching to the robot



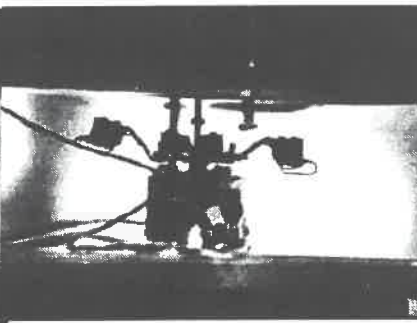
(2) Small robot succeeds in hanging on CMM



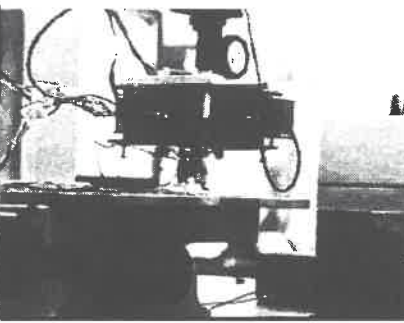
(3) It is transported to another working area



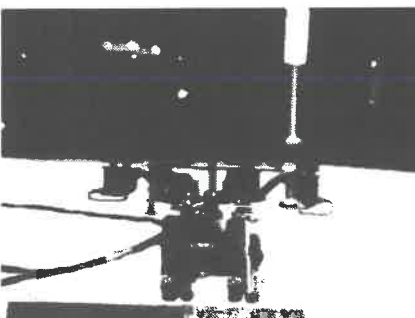
(4) Robot is released on the working surface



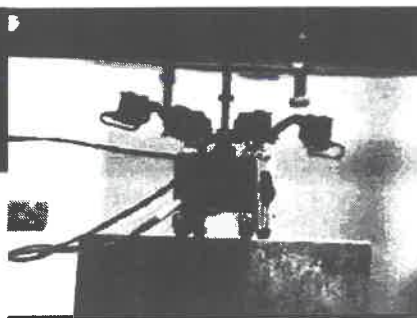
(5) It starts to sample the surface topography with tracking servo control



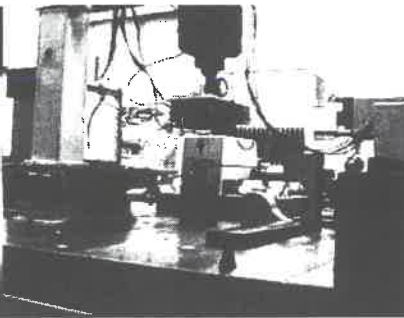
(6) CMM is capturing the small robot again after it finishes the task



(7) CMM can transport it even on the steep work



(8) After exploring the surface condition, it is released there



(9) They can stand by for the next task

Fig. 7 Cooperative action of CMM and micro robot by simple task-driven sequential programs

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Dimensional Metrology with the NIST Calibrated Atomic Force Microscope

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Because atomic force microscopes (AFMs) are capable of generating three dimensional images with nanometer level resolution, these instruments are being increasingly used in many industries as tools for dimensional metrology at sub-micrometer length scales. Sample properties commonly of interest in these applications include feature spacing (pitch), feature height (or depth), feature width (critical dimension), and surface roughness. To achieve high accuracy, the scales of an AFM must be calibrated. Available standards for this purpose are commonly calibrated using stylus instruments and optical techniques. The effectiveness of such approaches, however, can be limited by the differences in the working ranges of the various techniques and by questions of methods divergence. Such divergence may occur between measurements made by instruments using different techniques to measure the same feature. A reflected light microscope and an AFM, for example, may have differing sensitivity to the cross-sectional profile of measured lines. For sufficiently steep lines, an AFM tip will only contact the features near the tops. The non-uniformity of the line profiles could lead to a difference between optical and AFM pitch measurements.

To calibrate pitch, height, and possibly line width standards using an AFM, we have developed the calibrated AFM (C-AFM) at NIST. Our instrument has metrology traceable to the wavelength of light for all three axes. This is accomplished through the integration of a flexure x-y translation stage, heterodyne laser interferometers, and a digital-signal-processor based closed-loop feedback system to control the x-y scan motion. The z-axis translation is accomplished using a piezoelectric actuator with an integrated capacitance sensor, which is calibrated using a heterodyne laser interferometer. When fully developed, this instrument will be a calibration tool for pitch and height standards for scanning probe microscopes.

Recently, we have made significant improvements to our uncertainty budgets for measurements in all three axes. For example, a reevaluation of the tilting motions of the C-AFM scanner allowed a refinement in our estimate of the Abbe error contribution to our uncertainty in pitch measurements. During subsequent pitch measurements, we performed a test of this new model and obtained results consistent with our refined uncertainty budget. Our first pitch measurements for an external customer have now been completed. The samples measured included two types of patterns: a 0.6 μm grid and a 0.4 μm linear scale. Several improvements in the system, including the replacement of the z-stage, have improved the repeatability of $\sim 0.1 \mu\text{m}$ step height measurements by about a factor of three, and we have plans to perform our first step height measurements for an external customer in the next few months. We are also working toward the development of linewidth standards through the comparison of C-AFM width measurements with values obtained from other methods, including an electrical resistance technique. Our paper will describe the current status of the C-AFM, discuss the use of the instrument for measurements of pitch, height, and width, and describe our plans for future measurements.

Keywords: SPM, AFM, standards, metrology, pitch, scale calibration, step height, linewidth

Self-Contained, Six-Axis Positioning, Scanning and Metrology Stage with Active, Sub-Nanometer Planarity and Straightness Compensation

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Abstract

A novel actively-compensated six-axis stage has been developed with:

- Up to 200x200µm planar scan range
- High-throughput actuation
- Ångstrom-scale out-of-plane motion via integral, active, real-time compensation
- Fully internalized six-axis metrology; no requirement for bulky and costly external metrology such as interferometers with L-optics.
- Auxiliary vibration-cancellation technology which nullifies motion-driven resonances of adjacent componentry for enhanced throughput and resolution.

The self-contained, piezoelectrically-actuated stage, commercialized as PI Model P-915, leverages technologies originally developed for adaptive optical systems. It integrates eight servo-controlled PZT actuators and six diamond-machined capacitive sensors of a new design to reduce unwanted out-of-plane motions and rotational errors below 0.5 nm (RMS) and 0.1 arc-second, respectively.

The mechanism utilizes six capacitive sensors based on a newly-developed two-plate asymmetric design. The sensors consist of a probe plate and a slightly larger target plate. This configuration was developed to enhance linearity and insensitivity to the orthogonal axes' motions. The six target plates form a highly accurate coordinate reference frame. An advanced digital controller provides real-time compensation of orthogonal motion errors, utilizes scan-range-dependent loop gain settings, providing step/settle response <8msec independent of step size. It provides better than 0.033nm (0.33Ångstrom) RMS stability in the critical Z-axis.

Applications include current and emerging laboratory and industrial endeavors such as scanned-probe microscopy and pole-tip recession metrology. The ability of the system to actuate in a defined plane and correct for sample/fixtures coplanarity errors represents a significant advancement for processes requiring a variable absolute reference plane.

Active Compensation of Nanometer-Scale Motion Errors: Theory

In a single-axis actuation, there are three types of motion errors:

- 1) The misalignment between the measurement and motion axes introduces Abbé errors and cosine errors. The Abbé error is most significant since it scales with fixturing lever-arms. With reference to Fig. 1, it is:

$$\delta x_{Abbe} = y_{SX} \cdot \varphi_z - z_{SX} \cdot \varphi_y$$

- 2) The inevitable orthogonal motions introduces further measurement errors:

$$\begin{aligned} \delta x_{meas} = & \delta x + \frac{\partial f}{\partial y} \delta y + \frac{\partial f}{\partial z} \delta z + \frac{\partial f}{\partial \varphi_x} \delta \varphi_x \\ & + \frac{\partial f}{\partial \varphi_y} \delta \varphi_y + \frac{\partial f}{\partial \varphi_z} \delta \varphi_z \end{aligned}$$

$$\text{where } x_{meas} = x + f(\delta y, \delta z, \varphi_x, \varphi_y, \varphi_z)$$

- 3) Structural deflections within the sensor, or due to cable stresses, etc., cause additional deformations and motion errors which are impossible to model mathematically.

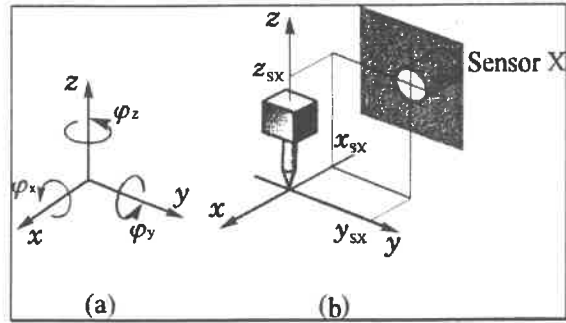


Fig. 1. The coordinate system.

These errors can be compensated via calibration only if they are repeatable. Usually they are not, and furthermore tend to vary significantly from system to system. Consequently, even closed-loop positioning systems exhibit hysteresis and unit-to-unit variability. Cabling and fixturing stresses add further uncertainty on the sub-nm scale.

To eliminate errors (1) and (2), simultaneous real-time metrology and loop closure in all six degrees of freedom ($X, Y, Z, \phi_x, \phi_y, \phi_z$) is required. It is less costly and more practical to do this than to attempt mechanical perfection or a predictive model which takes into account all the possible interrelating errors.

In the case of a raster scan in the XY plane, it is desired that zero motion occur in $Z, \phi_x, \phi_y, \phi_z$. Thus, ideally:

$$\delta x_{Abbe} = 0$$

$$\delta x_{meas} = \delta x + \frac{\partial}{\partial y} \delta y$$

The accuracy of this approach is limited only by the performance of the control system and displacement sensors. To realize a 6 axes closed-loop positioning system, PI developed a novel multi-axis flexure stage, a digital position controller and a two plate capacitive displacement sensor consisting of a PROBE and a slightly larger TARGET plate. This type of sensor was chosen because of its sub-nanometer resolution and insensitivity to lateral motion ($\frac{\partial}{\partial y}$ can then be neglected). Six TARGET plates form a coordinate reference block (Fig. 2). The sensors S_{x1} and S_{x2} measure the X-displacement and the rotation ϕ_z while the sensor S_y measures the Y-position. The sensors S_{z1}, S_{z2}, S_{z3} measure the z-position and rotations ϕ_x and ϕ_y .

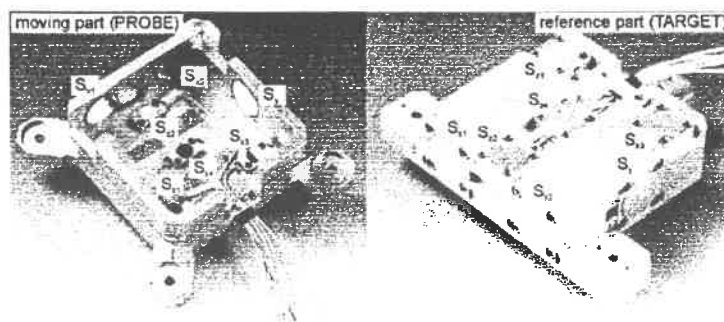


Fig. 2. Integral, simultaneous metrology of all six degrees of freedom.

Since the XY-plane and the YZ-plane are defined by more than one sensor, all TARGET electrodes in a plane are diamond milled in one production step. This ensures excellent orthogonality and manufacturability.

The digital controller provides flexibility and the computing power for digital filtering, linearization of the sensor signals and calculating the individual axis information from multiple sensor inputs as well as the individual PZT actuator drive signals. The digitized sensor signals are filtered and linearized with a 4th order polynomial

correction. The nominal range of XY and ϕ_z is $100 \times 100 \mu\text{m}^2$ and $\pm 500 \mu\text{rad}$, respectively. Fig. 3 shows the resulting Angstrom-scale out-of-plane motion throughout a $100 \times 100 \mu\text{m}$ scan.

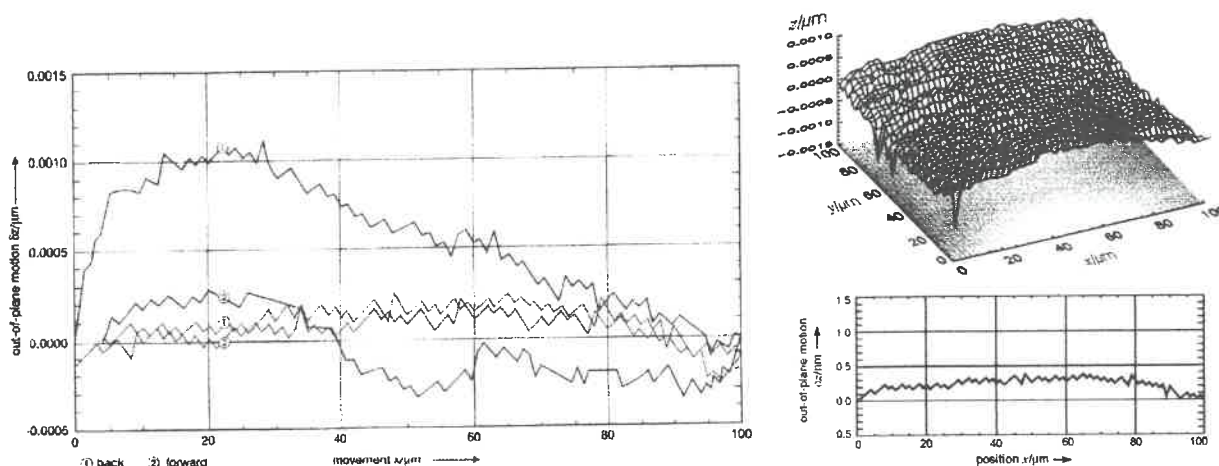


Fig. 3. Left: Single-axis, bi-directional trajectory control: passive (trace 1) and active (trace 2). Right: Out-of-plane motion (Z) over a $100 \times 100 \mu\text{m}^2$ scanning range

Throughput Enhancement by Vibration Nullification

In addition to the dimensional compression that has characterized the evolution of processes in the semiconductor, photonic, mass storage and biotech fields, economic considerations have forced a time compression as well. Settling phenomena now dominate the cycle times of many processes, and motion-generated ringing and resonance is commonly the most significant bottleneck to enhanced overall throughput and resolution. In collaboration with Convolv, Inc., PI has introduced a novel technology, trade-named Mach™, an implementation of the patented Input Shaping™ technology based on research performed at the Massachusetts Institute of Technology. This eliminates the ringing and resonance of the mounting frame and adjacent componentry caused by rapid actuation of the nanopositioning stage. Figs. 4 and 5 illustrate the benefits of this technology in step and scanning actuation, respectively. The Mach technology is implemented in a module which may be applied to virtually any PI closed- or open-loop system in addition to the P-915 six-axis actively compensated stage.

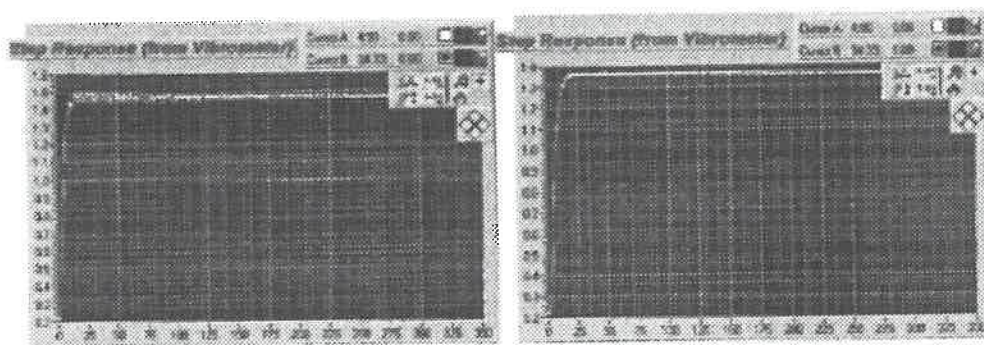


Fig. 4. Elimination of motion-driven ringing in the stage-mounted fixturing in rapid steps, as imaged by a non-contact Polytec laser vibrometer. Left: ringing is easily visible; settling takes several hundred msec. Right: Ringing is eliminated by use of Mach™ technology.

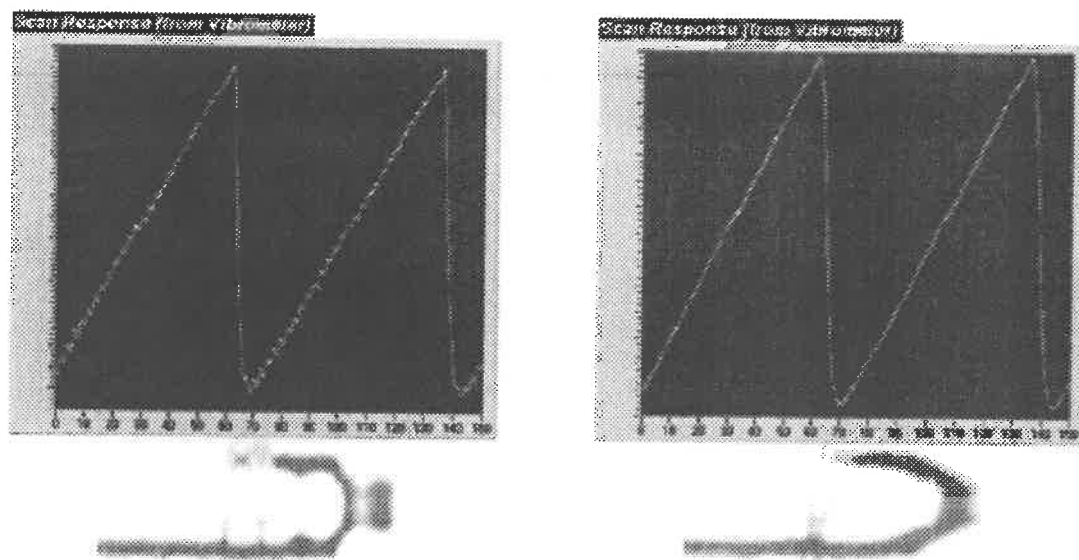


Fig. 5. Mach™ vibration nullification in a dynamical application: scanned metrology. Left: fixture resonance causes evident banding and smearing in image. Right: vibrations are eliminated (total image width $\sim 1\mu\text{m}$)

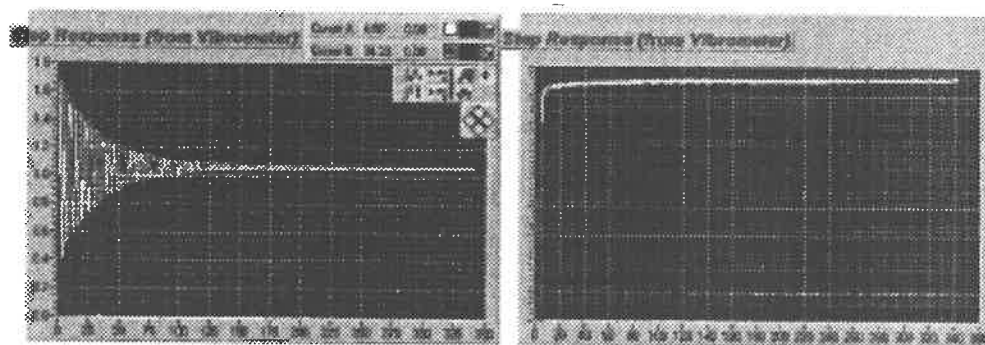


Fig. 6. Application of Mach™ vibration nullification to *open loop* nanopositioning.

Conclusion

These are enabling technologies for emerging surface inspection applications in disk drive and wafer manufacturing processes where an internal absolute coordinate frame reference is required and high throughput is a necessity. Immediate applications include near-field optical probing, waveguide metrology and fabrication, scanned-probe metrology, white-light interferometry and nanomachining. In particular, pole-tip recession metrology is highly dependent on an accurate and repeatable measurement reference plane and may represent the most significant near-term industrial application of this design approach.

KEYWORDS: active error compensation, sub-nm positioning, nanopositioning, nanoautomation, micropositioning, piezoelectric, active structure, out-of-plane, AFM, SPM, surface metrology, structured surfaces, pole-tip recession, microlithography, MEM, nanotechnology

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ACTIVE BEARING PRELOAD CONTROL OF A HIGH SPEED SPINDLE USING PIEZOELECTRIC ACTUATORS

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ABSTRACT

This paper will report an active bearing-preload control mechanism using piezoelectric actuators. In order to overcome the problems of nonlinearity, hysteresis and DC drift of the piezo-actuator, a strain gage load cell is integrated into the piezoelectric actuators. Bearing preload is monitored on line by the load cell and then fed back to the piezo-actuator. Results of the experiments conducted on a bearing testing stand are also reported. Test results show that the piezo-actuator can control the bearing preload well. The advantages of piezoelectric actuators are compact size, easy control, and precise motion. It can be easily implemented into a spindle without the increasing of the spindle size.

Keywords: high-speed spindle, bearing-preload control, piezoelectric actuators

INTRODUCTION

Angular ball bearings are the most popular bearing type used in the high-speed spindle. Proper preload of the angular ball bearing is important to the rigidity, accuracy and life of the spindle. Preload methods of angular ball bearings can be classified into two categories: constant offset preload and constant force preload. Constant offset preload spindle has the advantage of high rigidity. It is widely used in heavy cutting situations such as rough milling, tapping and drilling. However, the constant offset method is not favored in high speed cutting applications, because the preload could increase rapidly in high rotation speed due to the temperature-induced expansion and centrifugal force. The increased bearing preload, if not compensated or controlled, could cause shorten bearing life or even bearing seizure.

In many high speed cutting applications such as light milling and grinding, constant force preload method using spring force is popular. Constant spring force preload method can avoid the problems caused by thermal expansion and centrifugal force in high speed applications. However, the spindle rigidity is sacrificed. It is not suitable to heavy cutting conditions. Recently, the research has been focused on developing a wide-range-cutting spindle with adjustable bearing preload for the applications such as machining centers. Due to the variety of cutting processes of a high speed machining center, the bearing preload should be adjusted according to the cutting conditions. High preload is recommended in the case of heavy cutting in order to increase the spindle rigidity. As the rotation speed goes higher, the preload should be reduced in order to compensation the increased bearing preload due to thermal expansion and centrifugal force.

This paper will report an active bearing-preload control mechanism using piezoelectric actuators.

PIEZO-ACTUATOR AND LOAD CELL

The active bearing-preload control mechanism using piezoelectric actuators is shown in Fig. 1. A PZT piezoelectric actuator with a strain-gage load cell is implemented in an angular ball-bearing set with DB(back-to-back) arrangement. Requirement of the adjustable preload of spindles used in machining centers is 20kgf ~ 80kgf from light preload to heavy preload. Correspondent bearing offset is from several micrometer to 20 μ m. Therefore, stacked type PZTs with 15mm length and 10mm diameter are considered to be suitable for this project. Stacked type PZTs have the advantages of high stiffness and high expansion coefficient (could be as large as 0.15%).

In order to overcome the problems of nonlinearity, hysteresis and DC drift of the piezo-actuator, a load cell using strain gages is integrated into the piezoelectric actuators as shown in Fig.2. The load cell is built from four self-temperature-compensation strain gages which consists of a full-bridge of Wheatstone bridge. Fig. 3 shows the calibration result of the load cell. It shows that the load cell has very good linearity and sensitivity or S/N ratio.

EXPERIMENTAL RESULTS

A testing stand has been built to evaluate the bearing pre-load control using the piezo-electric actuator. The testing bearing is FAG HS7010. Fig.4 shows the test result when a constant offset preload with a 13kgf initial preload has been applied. The bearing temperature was risen from 26°C to 30.5 °C after a period 2 hours of a constant 2600 rpm. At the same time, the bearing preload was also increased to 16kgf. After that, the spindle speed was changed to 5500rpm and maintained at another period of 2 hours. The bearing temperature was risen to 42 °C and the preload was increased to 27.5kgf. Then, the spindle speed was changed to 9000rpm. The bearing temperature was risen to 48.5 °C and the preload was increased to 37kgf.

Test has also been conducted to investigate the effect of the initial pre-load to the bearing temperature. Fig. 5 shows the test result at a 4000rpm. The bearing was initial preload at 13kgf and reached to a steady value of 18kgf after 1.7 hours. After that the preload was suddenly increased to 40kgf. It shows that there is little effect on the bearing temperature. Fig. 6 shows a similar test at the condition of 8000rpm. From the test result, it can be concluded that the major effect on the bearing preload and temperature is due to the rotation speed but not initial pre-load.

Fig. 8 shows another test when active bearing pre-load control has been applied. The bearing was initial pre-loaded at 13kgf and operated at 2600rpm. After 1.6 hours, the spindle speed was changed 5600rpm for another period of 1.6 hours. The test result shows that the bearing preload has been maintained at 13kgf by the actuation of the piezo-electric actuator.

CONCLUSIONS

An active bearing-preload control mechanism using piezoelectric actuators has been demonstrated. This mechanism has the advantages of compact, easy control and fast response. It can be easily implemented into a spindle without the increasing of the spindle size.

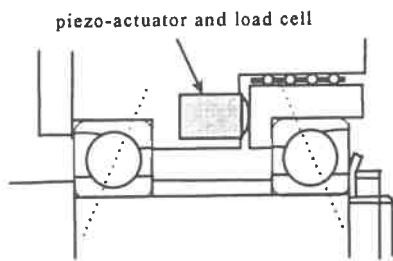


Fig. 1: bearing pre-load control

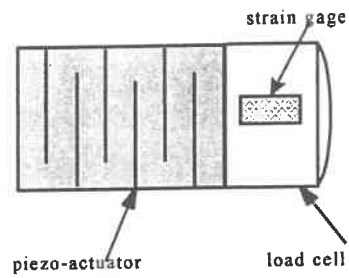


Fig. 2: integrated piezo-actuator and load cell

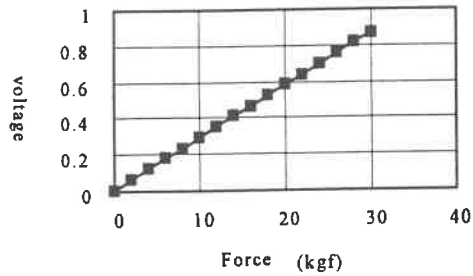


Fig. 3: calibration of load cell

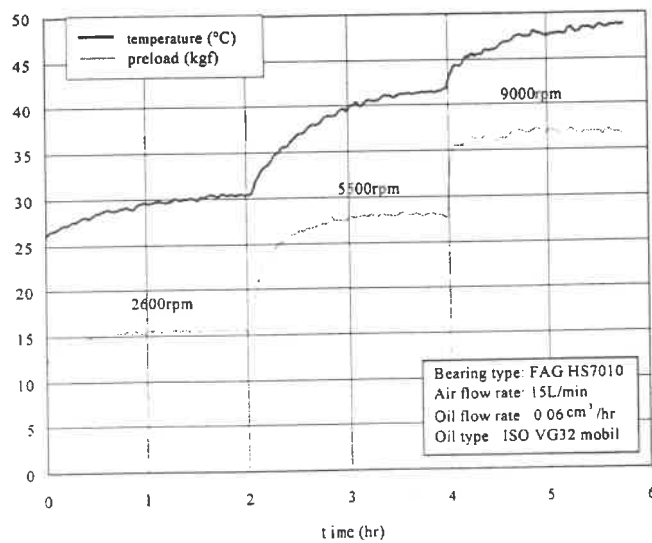


Fig. 4: Effect of spindle speeds on constant offset preload

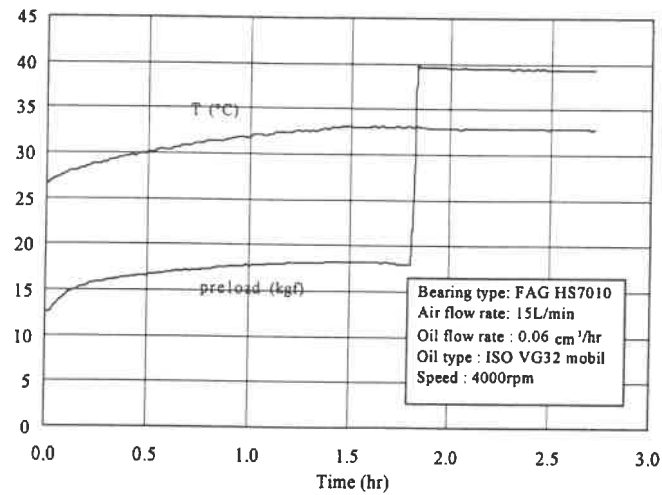


Fig.5: Effect of initial preload at 4000rpm spindle speed

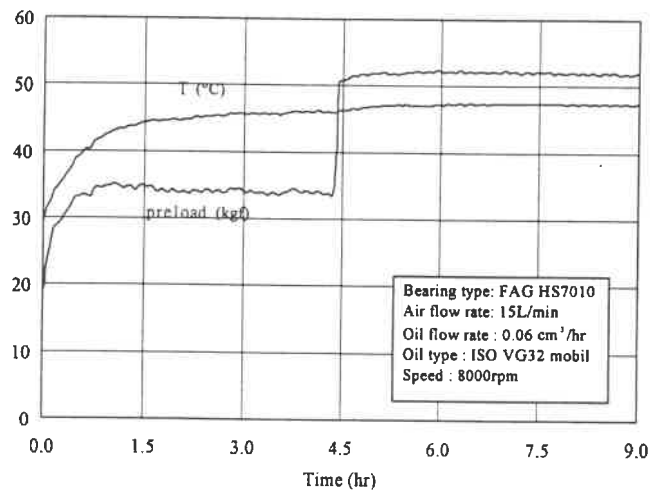


Fig.6: Effect of initial preload at 8000rpm spindle speed

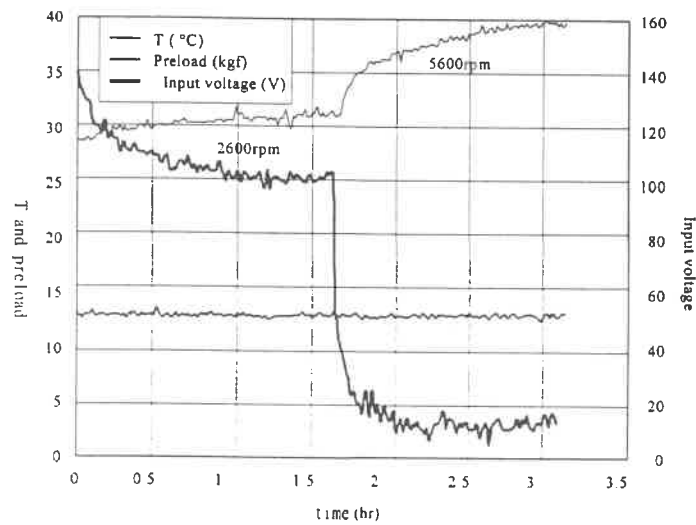


Fig. 7: Active bearing preload control

Development of a Nanometer Resolution Constant Velocity Piezoelectric Actuator

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ABSTRACT

The type of actuator described in this paper can provide constant velocity motion with extremely small following error (less than 20 nm). However, the speed of translation is limited by the size, and therefore the stroke, of the piezoelectric actuator. Two designs have been developed, one with a large high-voltage actuator (25 mm long, 25 μ m stroke at 1000v) and the other with a miniature low-voltage actuator (4 mm long, 4 μ m stroke at 60v). The maximum speed capability is a direct function of the stroke of the actuator, the expand/retract frequency of the piezoelectric actuators and the natural frequency of the structure. Since the natural frequency of the structure will be on the order of 500 Hz, the maximum operating frequency should be less than half of that value to avoid large structural vibrations and therefore large following errors. Therefore, the limiting speed can be estimated as:

$$\text{Max Speed} = \frac{500}{2} \cdot \text{stroke}$$

For the large actuator the maximum speed from this expression would be on the order of 6 mm/sec and 1 mm/sec for the smaller one. While not measured at the theoretical limit, the two actuators behave as expected and the larger one was operated at 3 mm/sec with a ± 2 μ m following error and the smaller one operated up to 0.3 mm/sec with a similar following error.

INTRODUCTION

An actuator that combines long range motion, nanometer resolution, and constant velocity (CV) is attractive for many industrial uses. Currently, designs such as the Burleigh Inchworm™ [1,2] achieve nanometer resolution and long range motion, but are limited to discontinuous stepping motion. Designs of this type of mechanism center around a single extension piezoelectric element with clamping piezoelectric elements at both ends. To achieve nanometer resolution and constant velocity, a new mechanism concept has been proposed which uses a pair of extension actuators and a pair of clamping actuators. The concept is based on a walking type motion as opposed to the stop and go motion associated with the Inchworm or stepper motors. By separating the central extension element into independent actuators, individual control of each clamp is possible, which allows smooth transition between clamp changes. Figure 1 shows the sequencing of the piezoelectric actuators for a classic Inchworm (left) and the constant velocity actuator [3] (right).

MECHANICAL DESIGN

One of the most difficult aspects in the design of this type of actuator is the clamp. The problems are related to a lack of clamping force and/or a 'glitch' (unwanted motion in the direction of travel due to clamp operation) that occurs for each clamp cycle. An attractive feature of piezoelectric actuators is their high stiffness. Typical unit stiffness values of 800 N-mm/ μ m are easily obtained. Due to their high stiffness, these actuators are capable of producing very large forces. The maximum force only occurs in the ideal case where the outer boundaries are infinitely rigid. In reality, the force produced by the actuator is much less due to the compliance of the surrounding structure. Assuming a linear system, the force generated by the piezoelectric actuator can be expressed as

$$F = \frac{(K1 \cdot K2 \cdot Kp)}{(K1 \cdot K2 + K1 \cdot Kp + K2 \cdot Kp)} \cdot \delta(V) \quad (1)$$

* Currently at Pratt&Whitney, West Palm Beach, FL

** Currently at Corning, Inc, Corning, NY

where $K1$ and $K2$ are the boundary stiffnesses, Kp is the stiffness of the piezoelectric actuators, and $\delta(V)$ is the displacement per volt of the piezoelectric actuator. Taking the limit of Equation 1 as $K1$ and $K2$ approach infinity results in

$$F = Kp \cdot \delta(V) \quad (2)$$

which is the ideal case. As is shown by Equations (1) and (2), the force is a linear function of the displacement and assumes the piezoelectric actuator is always in contact with the boundaries. In general, for a friction free clamp, there must be a finite gap between the piezoelectric actuator and boundaries when the clamp is off. The length of the gap is typically determined by the tolerances between the contacting surfaces. Therefore, to maximize the clamping force, the clamp design should maximize stiffness with minimal free expansion.

The clamp design shown in Figure 2 [4] functions by transmitting the horizontal force of the clamping piezoelectric actuator into a vertical force which grips the rod. For this design, the holding force of the mechanism was 15 N compared to the theoretical holding force of 500 N. The reduction in force was due to excessive compliance in the 45° rocker flexure. Figure 3 shows the same design in the vertical position[5]. In this configuration, the measured

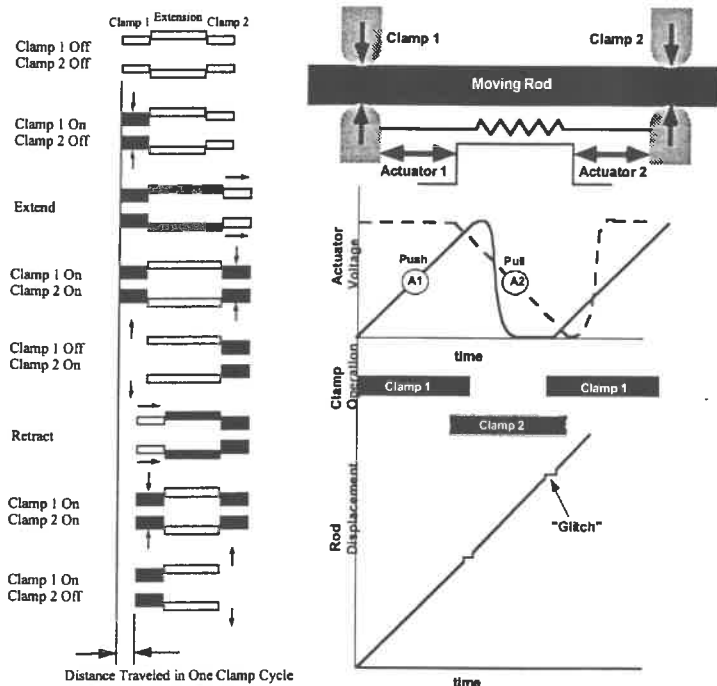


Figure 1: Piezoelectric Actuator Sequencing (Left) Classic Inchworm, (Right) CV Actuator

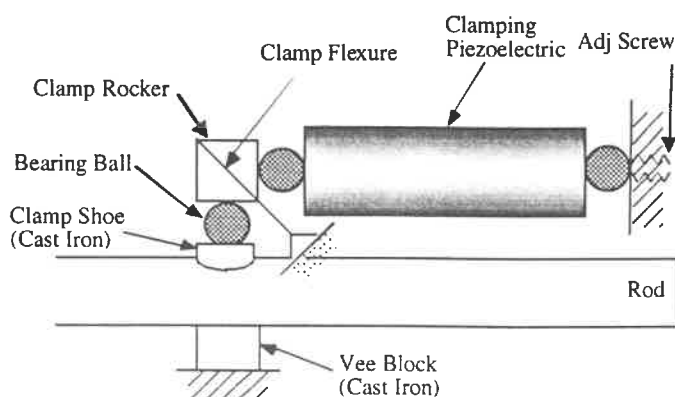


Figure 2: Clamp Design 1 in Horizontal Position

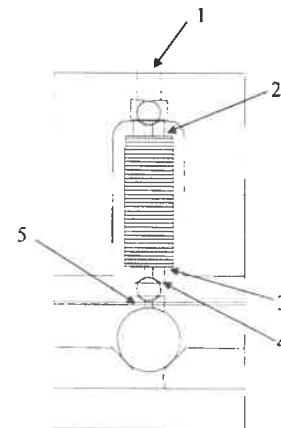


Figure 3: Clamp Design 1 in Vertical Position

holding force was 60 N. Again, compliance of the surrounding structure limited the maximum holding force. The geometry of the rocker flexure was one of the main sources of clamp glitch. As the piezoelectric actuator expands, the rocker element not only bends, but also expands. This results in motion along the axis of travel thus producing a glitch. In the vertical position, the glitch is reduced since the clamping action acts perpendicular to the direction of travel.

The clamp design [6] shown in Figure 4 uses a spring to provide the clamping force and the piezoelectric actuator is used to overcome the spring force and open the clamp. The main advantage of this design is the capability to have power off holding capability. To optimize the clamping force, the clamp beam and preload stiffness should be

selected to allow maximum motion of the piezoelectric actuator. The clamp beam stiffness was designed to be very stiff to minimize the relative deflection between the rod surface and piezoelectric actuator. Conversely, the preload spring must be as soft as possible to minimize the additional load on the piezoelectric actuator as the beam is lifted. This design was capable of generating a holding force of 65 N. The clamp glitch was measured to be around 200 nm and was caused by the angular position of the piezoelectric actuator with respect to the clamp beam. With proper assembly, the clamp glitch was reduced to 25 nm.

An entirely new clamp design was developed by Burleigh Instruments, Inc. for the Inchworm II™ actuator (US Patent 5,751,090). The clamp design shown in Figure 5 is referred to as an “inside-out” design. In this design, the moving rail is preloaded against the two clamp pairs and compensates for wear and temperature changes. The high tolerance fits required with the two previous designs are eliminated and only flat, co-planar surfaces are required. The clamp operates by energizing one pair of piezoelectric actuators and lifting the rail off the inactive clamp pair. This configuration eliminates residual friction forces between the off clamp and rail. The holding force is determined by the preload and coefficient of friction and is limited only by the deflection of the rails. To maximize the holding force, the preload force was applied equal distant from both clamp pairs which limits the maximum preload. Depending on preload and materials used, a holding force as high as 45 N was measured. The clamp glitch of the design was dependent on the geometry of the clamp flexure. The original flexure design was ‘L’ shaped which produced a glitch of 650 nm due to the unsymmetrical loading and deflection of the flexure (Figure 6). By modifying the design to be symmetric, the clamp glitch was reduced to less than 100 nm.

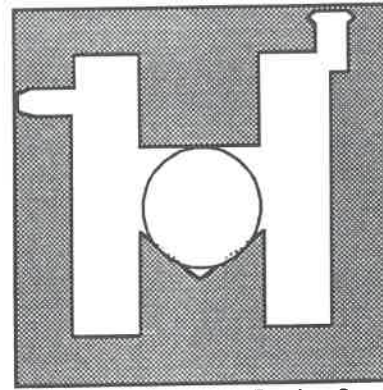


Figure 4: Clamp Design 2

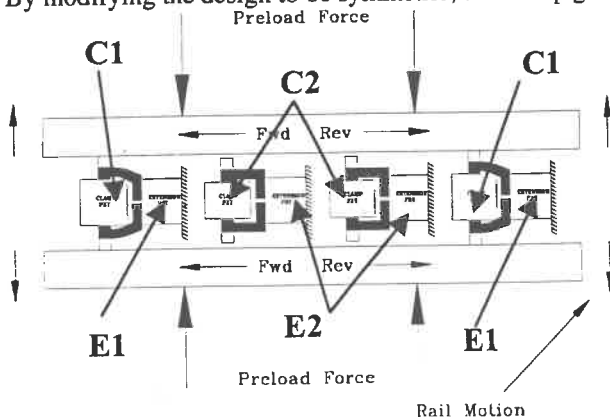
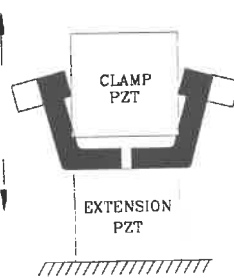
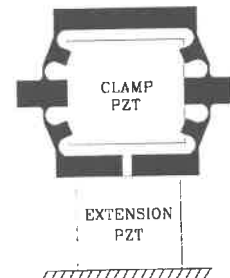


Figure 5: Clamp Design 3



'L' Flexure



Double Flexure

Figure 6: Clamp Design 3 Glitch

ACTUATOR CONTROL

Generation of the appropriate piezoelectric actuator control signals and the proper synchronization of these signals is essential to achieve constant velocity without disturbances. Since the constant velocity concept requires two extension and two clamping actuators, the generation and synchronizing of four simultaneous signals is required. For this reason, a high speed digital signal processor (DSP) has been employed. The waveforms used to produce constant velocity motion are shown in Figure 7. The extension waveforms shown in Figure 7 are for a push-push type of actuator. For a push-pull actuator, one signal would be inverted. The clamp actuators are synchronized to be on during the linear ramping portion of the corresponding extension actuator. For example, when clamp pair C1 is on, the extension pair E1 is extending linearly. When the clamps are off, the extension actuators retract with a half cosine function to provide a smooth transition to zero velocity at the end points. This reduces vibrations into the system. The beginning and end of the clamp waveforms are also half cosine functions to reduce vibration as well as protect the piezoelectric actuators from damage. The time variables which define the waveforms are based on the maximum extension actuator stroke and clamp overlap time.

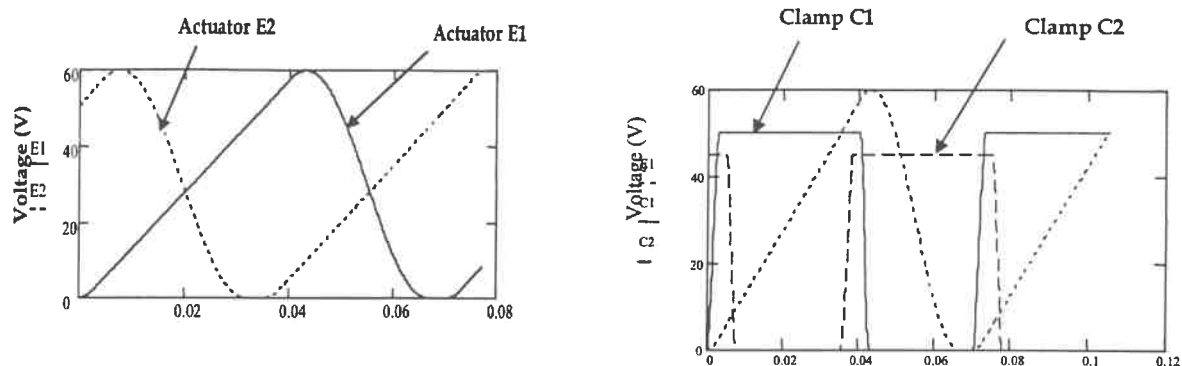


Figure 7: Extension Control and Clamp Control Signals

EXPERIMENTAL RESULTS

Both open and closed loop testing of the mechanisms were performed. The tests consisted of running the motors forward and reverse at various velocities and measuring the position using a capacitance gage. Figure 8 shows the measured response for open loop operation. In this Figure, the upper plot is the measured response in microinches and the lower plot is the input signal to the number 1 extension actuator in volts. A positive slope represents reverse motion and a negative slope represents forward motion. Subtracting the best fit line from the measured response results in the following error shown in Figure 9. The following error consists of two primary components: clamp glitch (discontinuous change in position) and PZT hysteresis (continuous non-linear motion between clamps changes). The asymmetric pattern in the following error is due to the hysteresis of the PZT. PZT material is typically more linear for increasing voltage (i.e. forward direction). This is clearly shown in Figure 9.

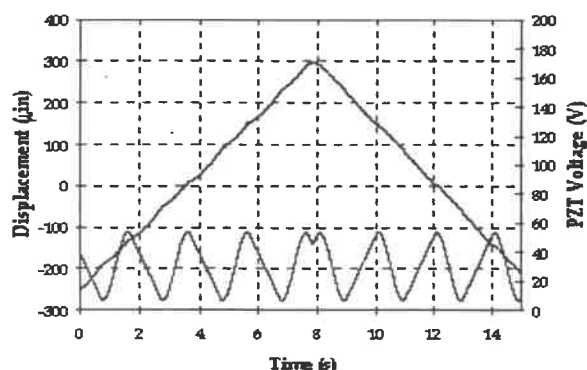


Figure 8: Open-Loop Response without Error Correction

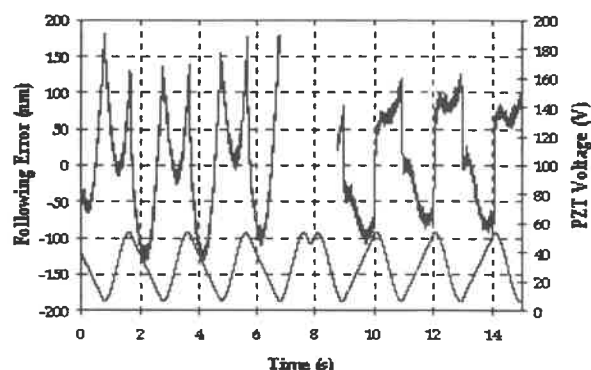


Figure 9: Open Loop Following Error without Error Correction

To improve the open loop response of the motor, a routine was written in the DSP code to automatically operate the motor and measure the clamp glitch and PZT hysteresis. Using this information, correction voltages are calculated and sequenced with the clamping and extension signals. Using this technique, the open loop following error was significantly reduced as shown in Figure 10. The peak-to-peak following error was less than 120 nm in both the forward and reverse directions.

For closed loop control, the capacitance gage was used as a feedback sensor for a PID controller. An initial set of PID gains were derived analytically to serve as a starting point. Experimental tuning of the PID gains was required to minimize the following error. Figure 11 is the following error with feedback and PID control for a velocity of 0.0015 mm/s. The peak-to-peak following error was less than 40 nm with the majority of the error being less than the sensor resolution.

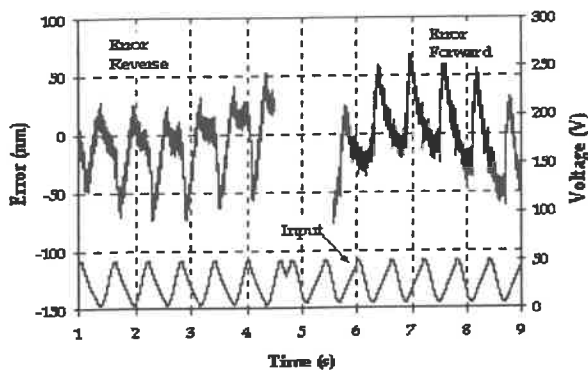


Figure 10: Open Loop Following Error with Error Correction

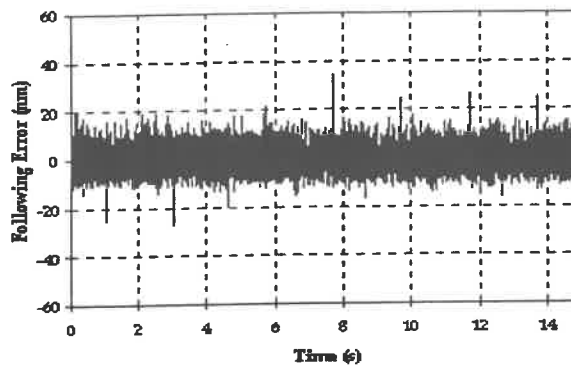


Figure 11: Closed Loop Following Error with PID Feedback Control

CONCLUSIONS

Three motors based on piezoelectric actuators have been developed which provide long range motion, nanometer resolution, and constant velocity. Primary error sources for mechanisms of this type were found to be inherent in the clamp designs and piezoelectric hysteresis. Design guidelines have been presented for developing a low glitch/high force clamping mechanism. Off-the-shelf DSP controllers have been used to generate the required control signals. These devices offer a flexible open architecture to develop numerous control schemes for both open loop and closed loop operation. Improvement in open loop response was demonstrated using a feed forward technique based on stored measured error data and resulted in following error less than 120 nm. Further reduction in following error was achieved using a PID controller with feedback. This resulted in a resolution greater than ± 20 nm for the entire range of travel of 12 mm.

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ENHANCING BALLSCREW AXIAL DYNAMICS

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A linear-motion system incorporating a leadscrew drive is shown schematically in Fig. 1. The leadscrew (1) is mounted to the machine base by means of rotary bearings (2) and (3) and driven by a motor (4) via a flexible coupling (5). The nut (6) is mounted to the carriage (7) which is constrained to move axially on linear bearings (not shown). In modern machine applications, such a system is usually operated under closed-loop control where the position of the carriage, the rotation of the motor, or both are measured and used as feedback (Chen and Tlustý, 1995). If the feedback signal involves only rotation of the motor, axial disturbances are rejected in proportion to the mechanical impedance of the leadscrew mechanism. If the feedback signal includes the position of the carriage, the achievable bandwidth of the mechanism is limited by its impedance. In either case, high mechanical impedance is a prerequisite for rapid, accurate motion of the system.

Prior Art

In designing a mechanism for high mechanical impedance, one usually seeks to raise its rigidity and damping while minimizing its inertia. For most configurations, the nut, support bearings, and coupling can be sized for high rigidity without significantly increasing inertia. But the leadscrew has longitudinal stiffness proportional to the square of its radius and rotary inertia proportional to the fourth power of its radius. Thus the tradeoff between stiffness and inertia drives the design toward relatively compliant screws.

During operation of the mechanism, friction between the screw and nut generates significant heat, leading to a temperature rise and thermal expansion in the screw. To accommodate this expansion, the screw is usually allowed to slide freely in the axial direction at the support bearing (3) furthest from the motor. Thus, the entire thrust load is transmitted to the base through the serial combination of the thrust bearing (2) and that portion of the screw between the nut (6) and thrust bearing (2).

The rigidity of the loaded portion of the screw is inversely proportional to its length, so that the overall rigidity of the mechanism decreases as the carriage moves away from the thrust bearing (2). For the representative system described in Table 1, we plot in Fig. 2 the response of carriage position to motor torque as a function of frequency for various carriage

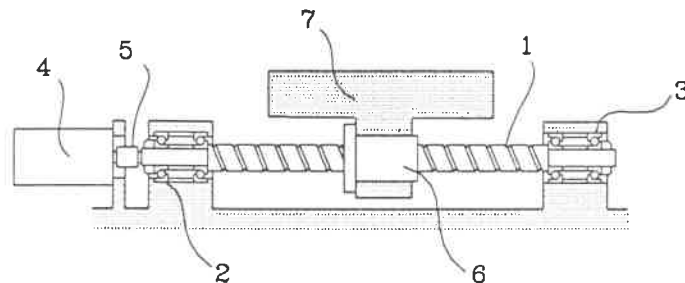


Figure 1: Schematic of a typical leadscrew drive system.

Table 1: Description of the system used in numerical examples.

leadscrew length	1 m
leadscrew average diameter	25 mm
leadscrew pitch	8 mm
leadscrew material	steel
thrust-bearing stiffness	500 N/ μ m
nut stiffness	600 N/ μ m
coupling torsional stiffness	150 kN·m
carriage damping factor	0.015
carriage mass	1000 kg
motor inertia	0.23 g·m ²

positions. As the carriage moves away from the thrust bearing, the resonant frequency drops and its peak amplitude rises.

The maximum bandwidth achievable with feedback proportional to carriage position can be determined graphically from Fig. 2 by drawing a line horizontally through the resonant peak, finding its intersection with the portion of the curve to its left, and reading off the frequency. A somewhat higher bandwidth can be obtained with proportional-derivative or more sophisticated feedback, but the resonant behavior of the leadscrew mechanism remains a limiting factor. It is therefore desirable to increase the resonant frequency and decrease the peak amplitude as much as possible.

An approach sometimes employed to increase rigidity is to use thrust bearings at both (2) and (3) and pre-stretch the screw in order to accommodate thermal expansion. Using this approach, the overall rigidity of the system can be increased, and the response of the system with the carriage near thrust bearing (3) is almost identical to that with the carriage near thrust bearing (2). The disadvantage of this approach is that it requires that a prohibitively high preload be applied in stretching the screw. For our example system, a preload of approximately 25000 N is required to accommodate a temperature rise of 20°C. Such a high preload increases the starting torque and heat generation in the rotary support bearings and can appreciably warp the bed of the machine.

To obtain a lower preload, a relatively compliant medium such as a stack of Belville washers can be inserted into the preload path at (3). The preload required to accommodate a given temperature rise drops directly with the stiffness of the preload medium, but so does the rigidity of the mechanism.

Damped Supports ¹

In the present work, it is proposed that a damping element be placed in the preload path at (3). For a relatively compliant but lossy preload medium, significant attenuation of the resonant peak amplitude can be obtained as shown in Fig. 2. In this case a damping washer

¹The method described in this paper for damping leadscrew vibration is subject to patent protection by the Massachusetts Institute of Technology.

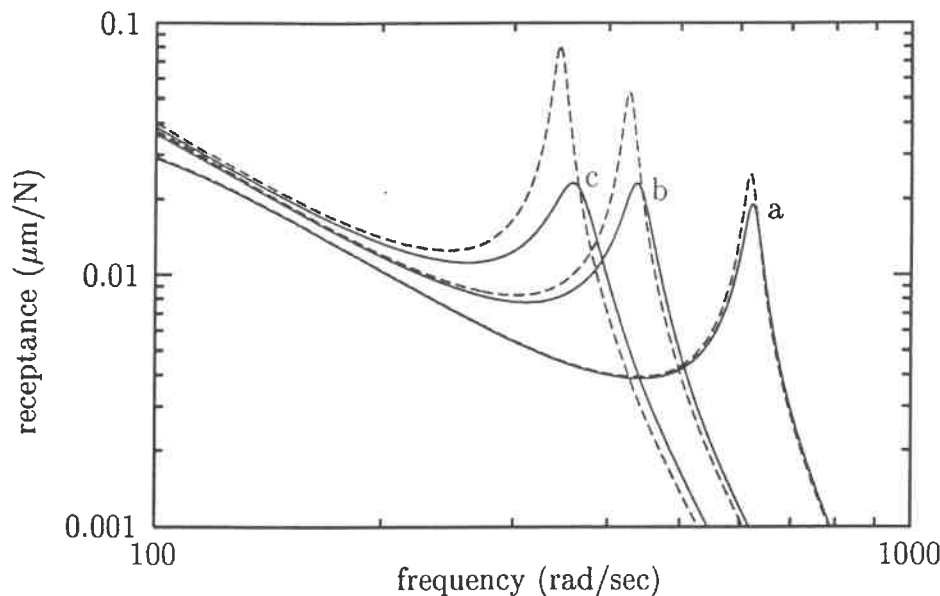


Figure 2: Comparison of the frequency responses obtained for the example system with the leadscrew unrestrained (dashed lines) and the leadscrew stretched against a damping element (solid lines); the carriage is positioned at (a) 5%, (b) 50%, and (c) 95% of its travel.

with $10 \text{ N}/\mu\text{m}$ stiffness and a loss factor of 1.0 is employed, and the preload necessary to accommodate a 20°C temperature rise is approximately 2500 N.

By use of a compliant preload medium, we only marginally increase the rigidity of the mechanism, and therefore obtain resonant frequencies only slightly higher than those obtained with the leadscrew end unrestrained. But we have greatly increased the damping, especially when the carriage is positioned near the damped support bearing. The result is that the bandwidth achievable with proportional feedback is nearly independent of carriage position.

Test-Bed Results

A carriage weighing roughly 190 kg and mounted on Thomson Accuglide size 35 linear guides was clamped to a shaft with a free length of 20 inches and a diameter of 0.625 inches. One end of the shaft was rigidly loaded against a pillow block and the carriage was positioned near the opposite end of the shaft. An accelerometer was mounted to the carriage to measure axial vibration, and the carriage was struck with an instrumented hammer. Measured receptances are shown in Fig. 3 for the cases where the second end of the shaft is free, preloaded rigidly against a pillow block, and preloaded via a viscoelastic washer against a pillow block.

Design Notes

Examining Table 1, we see that although there are a number of contributors to the compliance of a leadscrew drive, by far most compliant element is the screw itself. Moreover, if the lead is small compared to the diameter, torsional wind up of the screw can be neglected in

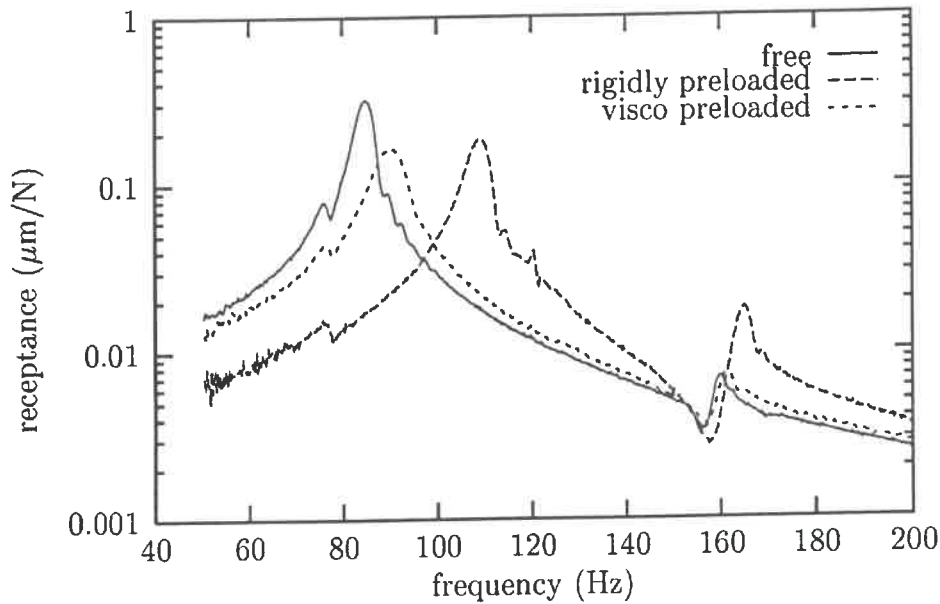


Figure 3: Measured receptances of the test slide with the shaft end free, preloaded rigidly against a pillow block, and preloaded via a viscoelastic washer against a pillow block.

comparison with its axial deformation. We are usually concerned with the performance of the drive over its entire range of travel, and we have seen that the damping is most important when the carriage is at the end most distant from the drive motor.

In that configuration (if we treat the leadscrew as the only compliance in the system), we find that both the damping and stiffness increase monotonically with the stiffness k_v and the loss factor η_v of the viscoelastic preload medium. But the preload force P necessary to accommodate a given temperature rise ΔT also increases monotonically with the stiffness of the preload medium according to (Nayfeh, 1998)

$$\frac{P}{EA\alpha\Delta T} = \frac{\eta}{\eta_v} = \frac{k_v L}{k_v L + EA} \quad (1)$$

where η is the loss factor of the drive system, and α , EA , and L are, respectively, the coefficient of thermal expansion, the stiffness per unit length, and the length of the leadscrew. We in general choose the preload medium to have as high a stiffness as possible without requiring an unacceptably large preload force.

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Hologram Diffraction Grating of Glass with Triangular Grooves Fabricated by Sacrificial Mask FAB Etching

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1 Introduction

A hologram diffraction grating is recently one of the key technologies of an optical disk drive. The hologram diffraction grating has several zones consisting curved grooves. It divides and bends a read-back beam to make several focus spots on optical sensors without a beam splitter or a convex lens. A *plastics* hologram diffraction grating mass-produced by injection molding for a DVD (digital video disk) has *square* grooves. To improve the read-back efficiency, the industry requires a *glass* hologram diffraction grating with *triangular* grooves. It is, however, very difficult to manufacture. The injection mold of the hologram diffraction grating with *square* grooves can be fabricated by conventional photolithography and RIE (reactive ion etching), yet that with *triangular* grooves needs an expensive fabrication method such as multi-exposing photolithography. *Glass* can be forged using a ceramics mold at about 700°C, yet the ceramics mold of the hologram diffraction grating can not be fabricated by cutting or etching.

In this paper, we introduce a new process to fabricate a grating with *triangular* grooves on a *glass* substrate. Figure 1 illustrates the fabrication process. First, we make a metal master mold of the grating with triangular grooves by mechanical cutting. Secondly, we make a plastics sacrificial mask with the same shape as the master mold on a glass substrate by pressing the master mold onto a plastics film. Lastly, we etch away the sacrificial mask using an FAB (fast atom beam)[1]. The sacrificial mask is etched and becomes thinner keeping the shape of the triangular grooves. When the sacrificial mask is completely etched away, the triangular grooves are reproduced on the glass substrate. The process reproduces a micro-structure (i.e. the triangular grooves) from the metal mold to the plastics sacrificial mask, and from the sacrificial mask to the glass substrate. The process contains two reproduction methods: forging of plastic and anisotropic dry etching of plastic and glass.

In this paper, we describe an experimental result in an early stage, and demonstrate that the process using only the inexpensive reproduction methods is promising for micro-reproduction of 1 μm features.

2 Experimental method

First, we make the master mold of a grating with saw-tooth-like triangular grooves of 1-3 μm in pitch. The grooves are cut on a 2×2 mm Ni plated aluminum block with a micro cutting machine. We designed the micro cutting machine shown in Fig. 2. It has a 2-axis coarse

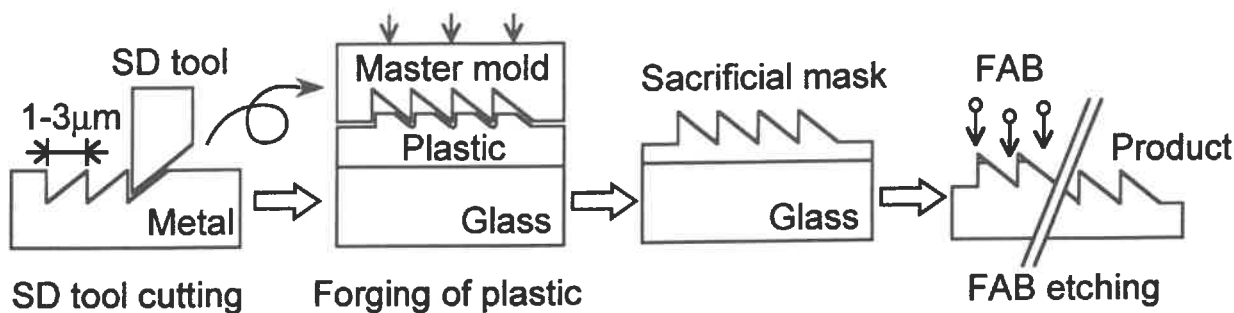


Fig.1 Schematic of the reproduction process of sacrificial mask FAB etching

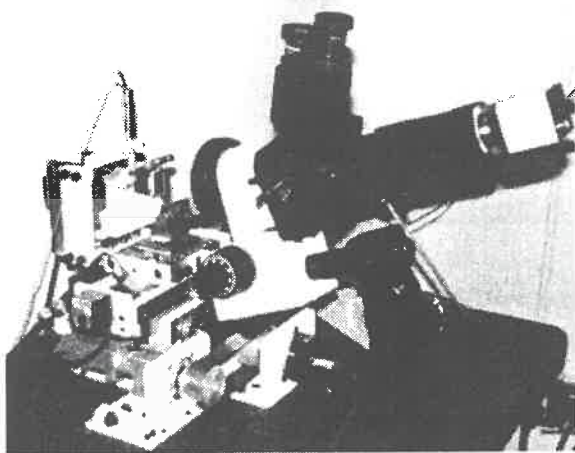


Fig. 2 Photograph of a micro cutting machine

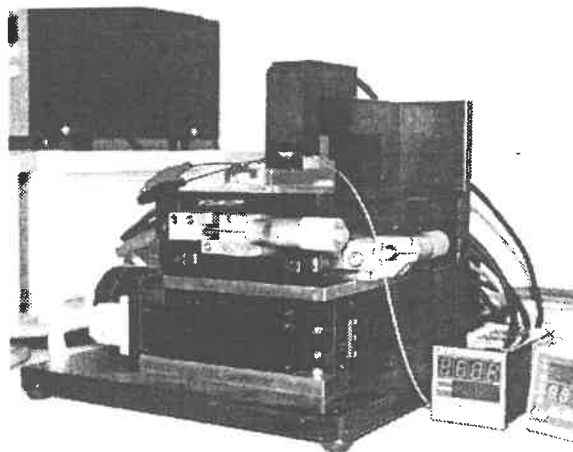


Fig. 3 Photograph of a micro press machine

translation mechanism using ball screws for longitudinal and transverse feeding, a 2-axis precise translation mechanism using piezo elements feedback-controlled by electric capacity displacement sensors for transverse and vertical feeding, and a single crystal diamond tool. The valley level of the grooves on the master mold should be within a 50 nm fluctuation, because the grating should be flat within a 50 nm fluctuation to prevent a wavefront aberration. The piezo elements compensate for the thermal deformation of the column and the fluctuating motions of the ball screws[2].

Secondly, we make a plastics sacrificial mask with the same shape as the master mold on a glass substrate by coating the substrate with a 1-3 μm thick resist and pressing the master mold onto the resist. We designed a micro press machine shown in Fig. 3. It is equipped with a force sensor and heaters for baking the resist. We press the master mold onto the resist at room temperature and 10-50 MPa.

Thirdly, we dry-etch away the sacrificial mask by an FAB. Our laboratory have developed an FAB source. Figure 4 illustrates the structure of the FAB source. It generates a plasma, accelerates an ionized gas by a 2-3 kV discharge voltage, and neutralizes ions in a carbon cathode. We set a workpiece on a stage which rotates at about 2 r.p.m. and is cooled at about -30°C . We etch the workpiece for 1-4 hours, and observe it with an SEM (scanning electron microscope).

3 Results and discussion

Figure 5 shows an SEM view of the master mold before being pressed onto a resist. We cut grooves twice per groove, that is, cut coarsely and finely at a cutting speed of 1500 $\mu\text{m}/\text{sec}$, and made about 3000 triangular grooves of 1-3 μm in pitch in about 6 hours. The surface of the grooves are smooth without any burrs. A *plastics* grating fabricated by injection molding using the master mold has a high read-back efficiency of about 90 %. It demonstrates that the grooves of the master mold have a nearly ideal saw-tooth-like triangular shape. We pressed the master mold onto several kinds of resist. Figure 6 shows an SEM view of the master mold after being pressed onto a resist. A crack was caused by the pressing force, because the master mold is made of a Ni-plated aluminum block (i. e. a hard Ni layer on a soft aluminum). In this paper, we use the crack to determine the place of the grating, and compare the shape of corresponding grooves on the master mold, the sacrificial mask and the glass substrate.

Figure 7 shows an SEM view of the sacrificial mask forged using the master mold shown in Fig. 6. We spin-coated the silica glass substrate with a 1.5 μm thick positive electron-beam resist (Nippon Zeon Co., Ltd. ZEP-520), and pressed the master mold onto the resist at 25 MPa and 25 $^{\circ}\text{C}$ without baking of the resist. Saw-tooth-like triangular grooves are reproduced accurately and sharply (more than that of the injected grating) from the master mold to the sacrificial mask.

Figure 8 shows an SEM view of the silica glass substrate etched by an FAB. We etched the workpiece by the FAB for 2 hours on condition that a gas of SF_6 was supplied to the FAB source at 25 SCCM, a discharge voltage and current were 2.5 kV and 50 mV respectively, a solenoid generated a magnetic field of 350-450 gauss in the FAB source, and the workpiece rotated at 2 r.p.m. and was cooled at about -30°C . The etch rate of the resist and the silica glass were almost equal and about 1 $\mu\text{m}/\text{hour}$. Saw-tooth-like triangular grooves are reproduced accurately from the sacrificial mask to the glass substrate, although the surface of the grooves is rougher than that on the sacrificial mask. The rough surface of the groove on the silica glass is caused by the rough surface of the resist exposed to the FAB. We think that the resist consists of several kinds of chemical or molecule which have different etch rates, and becomes rough by exposure to the FAB. We used several kinds of commercially available resist in experiments, yet all the resists became rough by exposure to the FAB. Future works include the improvement of a resist and the optimization of the conditions of the FAB.

4 Conclusion

To fabricate a *glass* hologram diffraction grating with *triangular* grooves, we developed a new process using a sacrificial mask and an FAB. The process reproduces triangular grooves from a metal master mold to a *plastics* sacrificial mask by forging, and from the sacrificial mask to a glass substrate by anisotropic etching. We designed a micro cutting machine to fabricate the master mold, and a micro press machine to forge the sacrificial mask. We produced triangular grooves of 1-3 μm in pitch on a silica glass substrate in the process to confirm the possibility of the process.

We thank Dr. Masahiro Hatakeyama and Mr. Katsunori Ichiki of Ebara Research Co., Ltd. for developing the FAB system.

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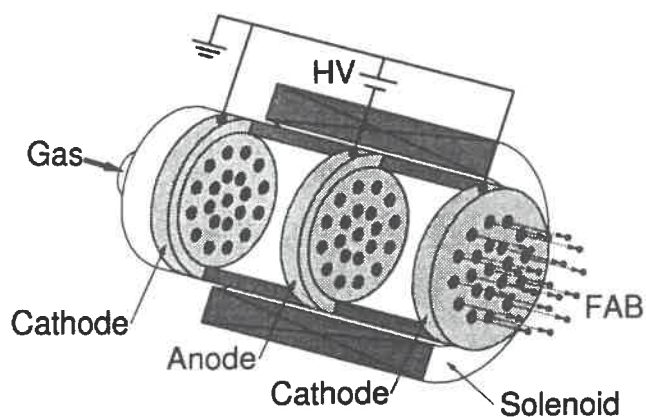


Fig. 4 Schematic of a fast atom beam source

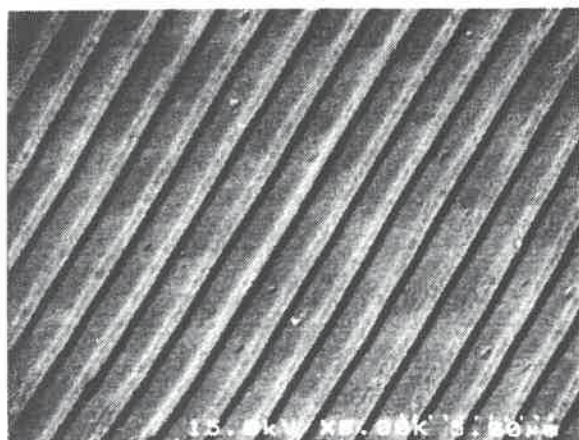


Fig. 5 SEM view of a metal master mold

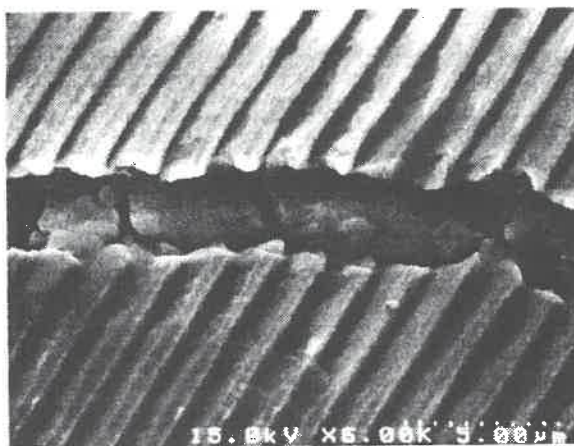


Fig. 6 SEM view of a master mold after pressing

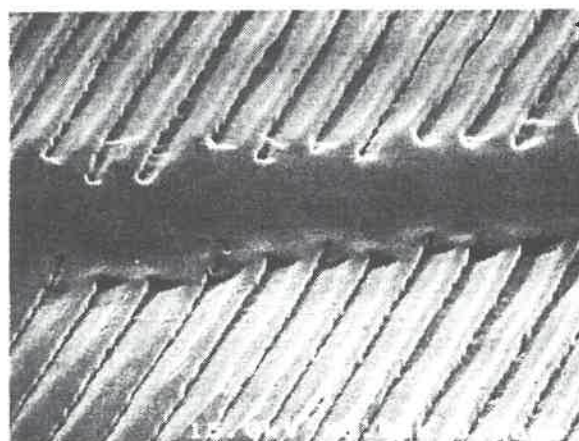


Fig. 7 SEM view of a plastics sacrificial mask

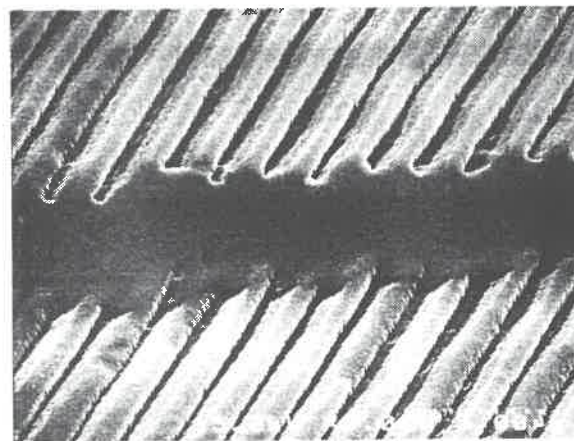


Fig. 8 SEM view of an FAB etched glass substrate

Wavelet Based Contouring Algorithm for Sub-Millimeter Diameter Optics

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Abstract

An ion beam micro-contouring process is being developed for producing sub-millimeter diameter optics. These optics include solid micromirror arrays and thin-film microelectromechanical (MEMS) mirrors having lateral dimensions less than 1 mm. While such optical components and arrays are finding increasing use in micro-photonic systems, no precise, controllable and robust process has yet been developed for their production. Ion beam machining is emerging as one promising alternative in this regard.

The contour etched by an ion beam will be equal to the convolution of the ion beam shape and an ion beam dwell function defined over an area of interest. We have developed a novel algorithm to model the deconvolution process, in which the desired removal contours and ion beam shapes are synthesized numerically as wavelet expansions. The ion beam shape is assumed to be fixed. The beam dwell function needed to produce the removal contour is then computed using a wavelet deconvolution. Experimentally, the ion machining is performed by rastering a 100- μm focussed ion beam generated by a duo-plasmatron ion source over the removal surface. Rastering is implemented using an orthogonal set of electrostatic plates interfaced to a computer guidance system.

In this paper, synthesized wavelet approximations of measured optical surface contours and subaperture ion beam material removal functions ("beam shapes") are mathematically combined to provide a prescribed "tool path" for modification of the existing optical contour to some other desired contour. The algorithm is implemented numerically and is compared to Fourier deconvolution techniques that historically have been used for such processing.

Problem Characterization

A convolution model has been used to characterize the ion beam machining process. In this model, the following assumptions have been made:

- 1) The beam function (the material removal profile of the ion beam) is fixed. It changes neither spatially nor temporally during the ion machining process.
- 2) Material removal is isotropic and proportional to dwell time.
- 3) The beam function is insensitive to position.

The ion machining operation can be expressed as

$$R(x,y) = \iint B(x-x_0, y-y_0) D(x_0, y_0) dx_0 dy_0 \quad (1)$$

The beam function can be thought of as the time the beam spends at a location on the substrate per unit area. For a given initial surface $g(x,y)$ and a desired final surface $h(x,y)$, the difference between the two yields the removal function $R(x,y)$. With a fixed beam function $B(x,y)$ determined by the physical parameters of the ion machining system, the time $D(x,y)$ over which the center of the beam must dwell at each point (x,y) can be computed by deconvolving $B(x,y)$ from $R(x,y)$.

The total workspace is broken down into square grids of unit area and the beam is centered at each location for a time approximately equal to the product of the computed dwell function at the square's co-ordinates (x,y) and the area of the square.

Thus,
$$\text{dwell time at } i,j = \iint D(x,y) dx dy \approx D(x_i, y_j) \Delta x_i \Delta y_j \quad (2)$$

This relationship allows us to express eqn. (1) in discretized form as follows:

$$R(x,y) = \sum_{i=0}^n \sum_{j=0}^m B(x-x_i, y-y_j) D(x_i, y_j) \Delta x_i \Delta y_j \quad (3)$$

This discretization of the removal function suggests that the figuring process can be discretized in a similar way thereby representing the removal process as a square discrete convolution.

Limitations of Fourier Deconvolution Techniques

Traditionally, Fourier techniques have been used^[1] to perform the deconvolution operation. Convolution of the two functions in the time domain can be expressed in the frequency domain as a product of their Fourier transforms, e.g.

$$R = B \cdot D \quad (4)$$

where, in this case, R = Fourier transform of the removal function;

B = Fourier transform of the beam function;

and D = Fourier transform of the dwell function.

Thus, in the Fourier implementation the computation of the dwell function simply involves dividing the beam function and taking the inverse Fourier transform of the result.

$$D(x,y) = F^{-1}\{R(k_x, k_y)\}B(k_x, k_y) \quad (5)$$

Fourier techniques are very successful in modeling periodic functions. Discrete Fourier transforms are periodic in nature and hence cause problems if used to transform non-periodic functions. In our case, both the removal and beam functions are non-periodic and Fourier techniques lead to incorrect results.

Also, the deconvolution step in eqn. (5) is unstable and has difficulties where the transform of the beam function approaches zero. This instability can best be explained by examining the case of a 1-D gaussian beam function. The Fourier transform of a gaussian function, for e.g. $B(x) = \exp(-x^2/\lambda^2)$ is itself a Gaussian $B(k) = \exp(-k^2\lambda^2/4)$, where k is the free variable in Fourier space and λ is the FWHM of the function. The inverse Fourier transform of this function represented by eqn. (5) is divergent because the denominator approaches zero.

Another problem in the Fourier approach arises from the discretization of the workspace. The computations, which are performed using discrete Fourier transforms, require that the inputs be periodic on finite rectangular or square arrays. Even though the metrology provides a rectangular or a square array of surface heights, the workspace itself typically will have a different shape, e.g. circular

or elliptical. Setting any portion of the data array outside the area to zero can lead to a bad spectral estimate of the complex spectrum corresponding to the optical surface.

The Wavelet Algorithm

We have developed a novel algorithm for the ion figuring process based on the series solution technique. In our algorithm, each element of the convolution process is expressed as a series of the derivatives of gaussian wavelets. The removal function, for e.g., can be expressed as

$$R(x,y) = \sum_{n=0}^{\infty} \sum_{m=0}^{\infty} c_{nm}^{(\lambda_x, \lambda_y)} \delta_{\lambda_x}^n(x) \delta_{\lambda_y}^m(y) \quad (6)$$

where $\delta_{\lambda_1}^n(x)$ and $\delta_{\lambda_2}^m(y)$ are the n^{th} and m^{th} derivatives of the gaussian in x and y ,
 $c_{nm}^{(\lambda_x, \lambda_y)}$ is the $(n,m)^{\text{th}}$ coefficient of the series expansion,
 λ_x, λ_y are the FWHM of the wavelet along the x and y axis respectively.

Similarly, the beam function can be written as

$$B(x,y) = \sum_{n=0}^{\infty} \sum_{m=0}^{\infty} a_{nm}^{(\lambda_x, \lambda_y)} \delta_{\lambda_x}^n(x) \delta_{\lambda_y}^m(y) \quad (7)$$

The series coefficients c_{nm} and a_{nm} are computed using orthogonality and orthonormality of the basis functions. The coefficients b_{nm} of the dwell functions are obtained by deconvolving a_{nm} from c_{nm} . Finally, the widths of the three functions (removal, beam and dwell) are related by

$$\lambda_R^2 = \lambda_B^2 + \lambda_D^2$$

Using the b_{nm} coefficients and the widths of the basis functions, the dwell function can be synthesized

as follows:

$$D(x,y) = \sum_{n=0}^{\infty} \sum_{m=0}^{\infty} b_{nm}^{(\lambda_x, \lambda_y)} \delta_{\lambda_x}^n(x) \delta_{\lambda_y}^m(y) \quad (8)$$

Simulations

The adjacent figures show the measured contour of a nine-element (350 x 350 μm^2) continuous mirror segment. This surface has a maximum (peak to valley) non-planarity of 200 nm.

Figure 4 shows a contour plot of the removal function needed to planarize the mirror surface. This removal function is to be ion machined by an ion beam having the beam function shown in Fig 5. The beam function is assumed to be axially symmetric (with a FWHM of 30 nm), although the algorithm can be used for any arbitrary beam profile.

Wavelet Deconvolution

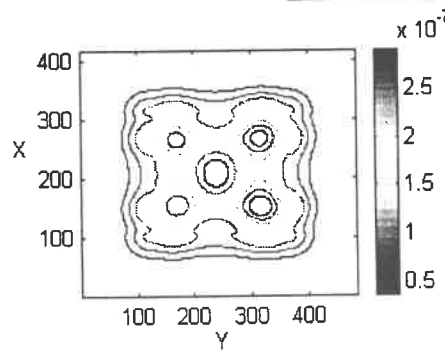


Fig 4 Desired Removal Contour

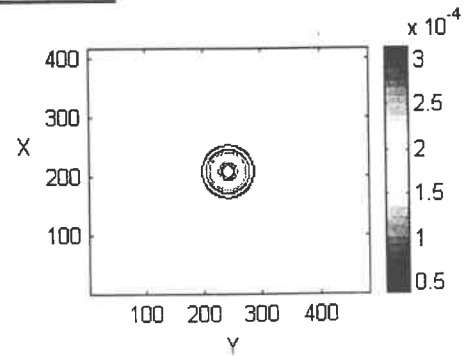


Fig 5 Measured Beam Shape

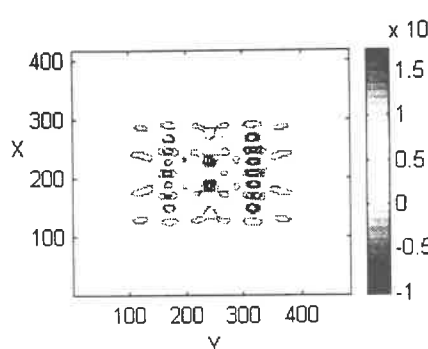


Fig 6 Calculated Dwell Time

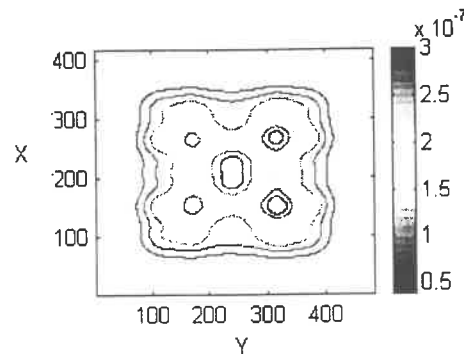
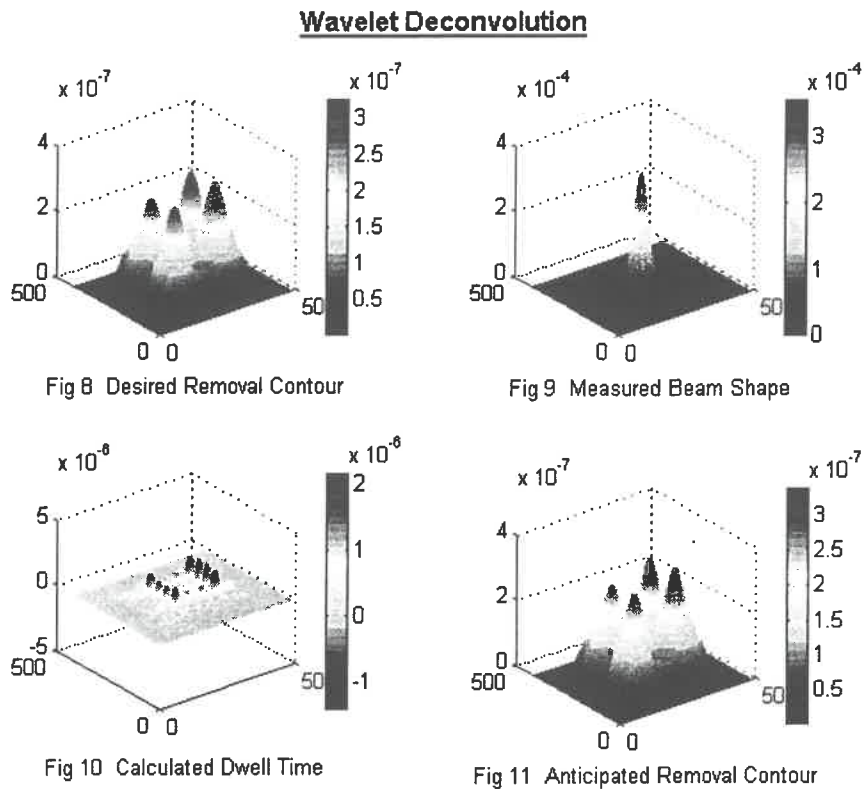


Fig 7 Anticipated Removal Contour

The contour plot of the calculated dwell time is shown in Fig 6. This function was obtained by deconvolving the beam function $B(x,y)$ from the removal map $R(x,y)$. The series wavelet expansion included 25 terms.

Fig 7 shows the anticipated removal contour map which will be removed from the mirror surface by the ion beam profile of Fig 5.

Figures 8 thru 11 show the surface plots of the elements in Figures 4 thru 7. These plots have been included for better visualization.



From these simulations we note the following:

- 1) The shape of the calculated dwell function $D(x,y)$ in no way resembles the beam function or the removal function. This result clearly indicates the ability of the algorithm to generate non-intuitive dwell functions;
- 2) Using wavelets, finely detailed structures can be produced on the ion machined surface even with a relatively wide beam.

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Machining beryllium spheres and shells for ignition capsule applications¹

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As part of an initiative to produce capsules for the National Ignition Facility (NIF) we established procedures for machining beryllium spheres and spherical shells with a diameter of 2.2 mm. This paper deals with the machining and finishing steps and the characterization done to date on the spheres. Details of bonding hemispherical cavities cut into beryllium cylinders are found in [2].

Whenever two sides of a part must be machined to high precision and registration, we use a reversible kinematic fixture called the Quick Flip (QF) [2] (Patent applied for) that reverses and relocates parts with a precision better than $0.1\text{ }\mu\text{m}$. For spheres, we begin with a cylinder of beryllium approximately 3 mm in diameter and 6 mm long. In the case of spherical shells, the starting point is a pair of 3 mm diameter by 3 mm long cylinders with a hemispherical cavity, a bonding surface, and a location surface machined in one end. Two of these cylinders are bonded together as described in [2] to give a 3 x 6 mm cylinder with a spherical cavity. A part holder (brass, generally) is machined to the sections shown in figure 1a, and screwed into the center of a QF locator and fixed into position with Loc Tite® adhesive. Permanent magnets hold the locator on the kinematic mount on the lathe spindle. The central bore 6 mm long is cut on the machine for a zero clearance fit to the beryllium cylinder. An additional 3 mm length bored in the same operation at slightly larger diameter aids the alignment. The centerline of the fixture is determined by measuring the part holder thickness. Then we press the cylinder into the brass part with hand pressure as shown in the left drawing in Figure 1 until it bottoms in the 6 mm 0-clearance section. The excess brass material (the lead length bore) is cut away from the part holder on a conventional Hardinge lathe (Figure 2.).

Precision lathe: kinematic spindle fixture.

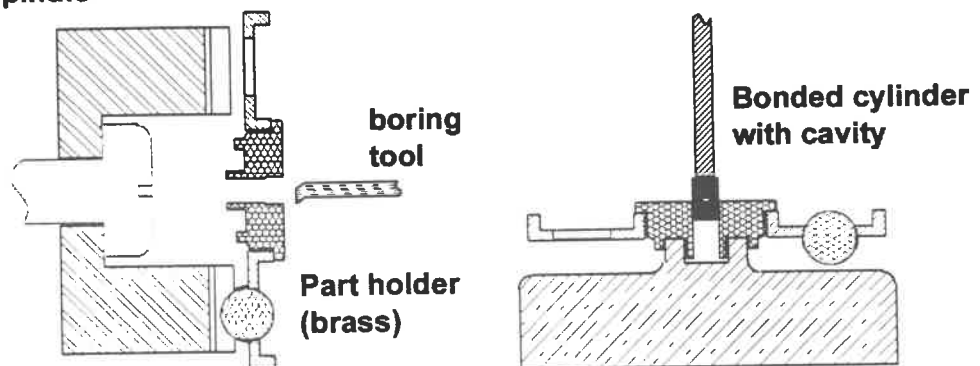


Figure 1. The quick flip fixture accepts the bonded spherical cavity. The left sectional view shows the brass part holder. The right shows the bonded cylinder being pressed in to the brass part holder.

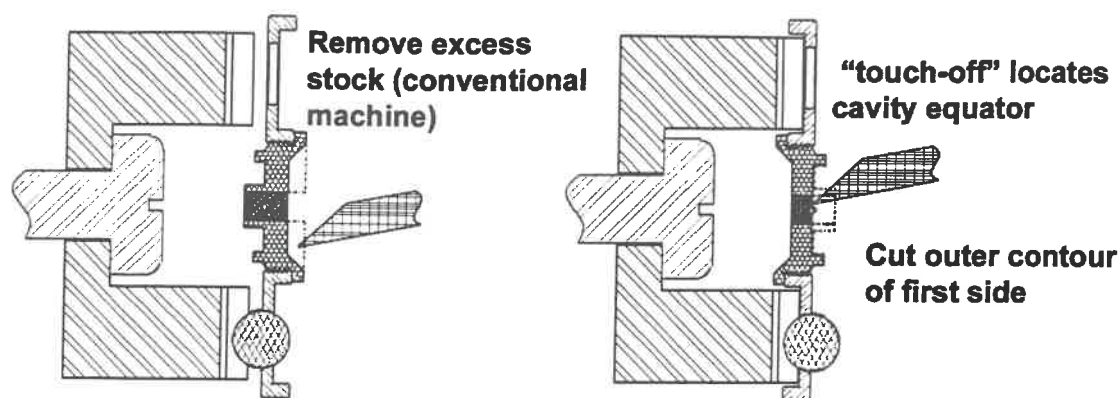


Figure 2. Removal of excess stock (essentially the lead cylindrical portion of the part holder), and cutting the first spherical contour (sectional views).

The reversible locator returns to the Pneumo precision air bearing lathe for the final machining of the spherical part (Figure 2., right sectional view). In the case of the spherical shells, the center of the cavity must be located. A touch off operation (the tool is brought in under computer control until it just touches the end of the beryllium cylinder) performed on both ends of the cylinder locates the cylinder ends relative to the centerline of the QF fixture, thus determining location of the equator of the cavity. In a roughing operation, the excess beryllium is cut away in a series of facing cuts that end at a radius larger than the outer spherical surface of the part. A pocket for the part support ring is cut in the brass stock. The final spherical radius is cut in Cartesian coordinates with the centerline of the tool at 45° to the axis of the lathe spindle (Figure 3.).

Glue support ring to
first surface and
part holder



Flip and cut 2nd
surface through to
support ring

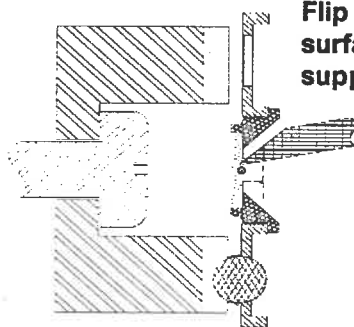


Figure 3. Attaching the support ring (right) and machining of the second spherical surface (left).

We remove the part from the lathe and epoxy a brass part support ring to the part holder and the beryllium spherical surface. This ring holds the part to the QF fixture when the other surface is cut. It fits into the pocket cut for this purpose in the part holder and includes a generous spherical cavity for the spherical surface just cut. The QF fixture returns to the lathe in lathe reversed position and the second spherical surface is cut in the beryllium in the same manner as the first side (Figure 3., left view). As this operation ends, the original part holder is cut through, and the support ring and epoxy hold the sphere to the fixture. To ease part removal we reverse the fixture again and cut through the part support ring to expose the epoxy. Dissolving the epoxy frees the part.

Scanning electron micrographs of a sphere cut by this method from S 200 D beryllium appears in Figure 4. The equatorial region is shown on the right. An slight groove appears in the magnified equatorial region in the left micrograph. The groove is somewhat larger than the machining marks (due in part to

the large grained beryllium stock) and further analysis on our Wyko Rough Surface Tester show that this groove can be a micron deep. Each spherical surface was cut 12 μm past the equator and this might be the reason for the groove.

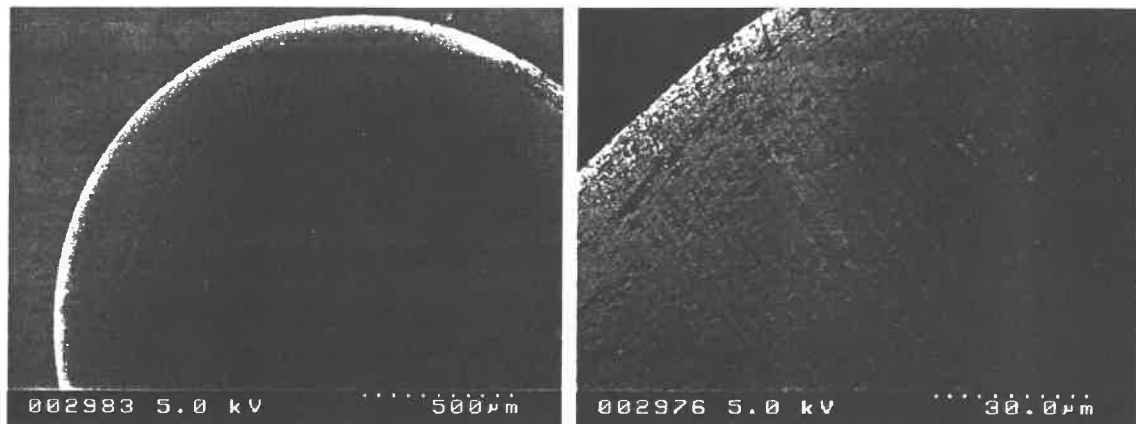


Figure 4. Scanning electron micrographs of a solid beryllium made by the method described in the text.

A spherical shell, a prototype of a NIF ignition capsule, was made from S200-D that was swaged at 400° C (see[2]) and exhibits a grain structure that is sub-micron. This material machined much better than the untreated material, exhibiting machine marks at ~ 100 nm rms over the surface. We gave this shell a quick polish in a variation of a ring lap. Both the bottom plate and the top plate have circular tracks machined in polishing pitch. The beryllium shell rotates in the intersection of an upper and lower polishing track. In operation, the bottom drive plate rotates about a center offset from the center of the polishing track and the top plate rotates about its own center. A 15 minute polish with 0.3 μm Baikalo polishing medium reduced the surface roughness to 10 nm rms. Figure 4 shows two contact x-radiographs [4] of a spherical shell made by the method described above. The views are approximately 90° apart in orientation. Image contrast enhancement shows a faint ellipse in each view due to residual copper left from the bonding process. Analysis of the images shows evidence of a machining burr ~ 9 μm long on the equator. Accounting for the burr, our estimate of the wall thickness varies from 163 μm by ± 3 μm ; the design thickness was 160 μm .

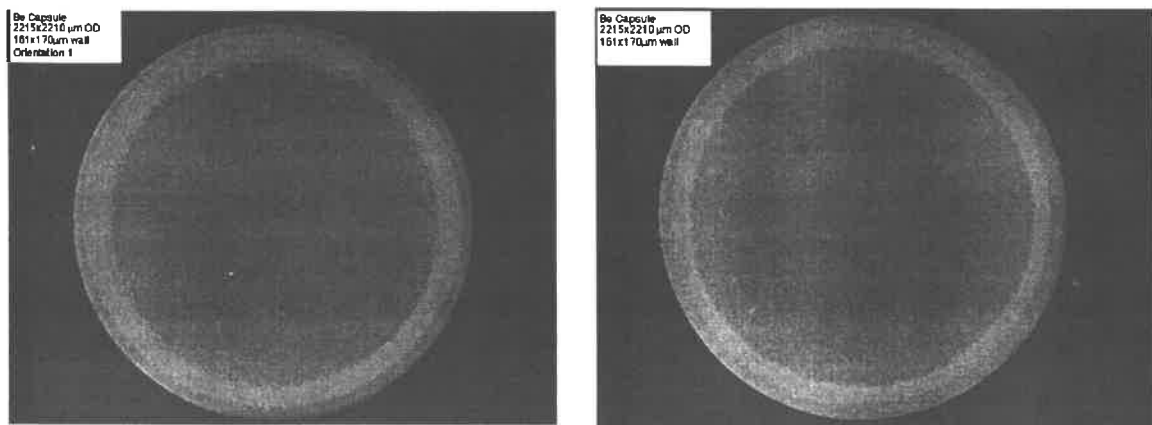


Figure 4. Radiographic views of a beryllium shell machined as described. The left view is approximately polar and the right is approximately equatorial. The equator can be seen as a faint ellipse with major axis from 1 o'clock to 7 o'clock in each view.[4]

[1] This work was supported by the U.S. Department of Energy under contract number W7405-Eng36.

[2] Margevicius et al., to be published in Fusion Technology.

[3] Salzer et al., Fusion Technol. **31** 477 (1997)

[4] We are grateful to our colleagues Martin Hoppe, David Steinman, and Richard Stephens at General Atomics, San Diego, CA for providing radiographic analysis of this shell.

The In-orbit Refocusing Mechanism for the Meteosat Second Generation Weather Satellites

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The Meteosat Second Generation (MSG) program is being implemented by the European Space Agency (ESA), in cooperation with the European Organization for the Exploitation of Meteorological Satellites (EUMETSAT). Prime contractor of the MSG program is Aerospatiale in Cannes (Fr). The Imager of the MSG satellites, called Spinning Enhanced Visible and Infra-Red Imager (SEVIRI), is developed by Matra Marconi Space France (MMS-F). The purpose of the REfocusing Mechanism (REM) is to provide a very accurate linear displacement capability for the secondary (M2) and tertiary (M3) mirror assembly of SEVIRI. The paper will describe the REM development and test results.

The main functions of REM are in more detail:

- Support the 3.9 kg M2/M3-mirror-assembly (CoG 110 mm above REM) during (ARIANE 5) launch **without a launch locking device**.
- Provide a controlled linear motion of the M2/M3-mirror-assembly for a stroke of ± 1 mm (pointing requirements: lateral deviation/decenter < 25 μm ; spurious rotations/tilt < 25 μrad).
- Maintain the M2/M3-mirror-assembly in a fixed position over the lifetime (stability requirements: axial deviation < 10 μm)

The development of the MSG REM is driven by stringent requirements on pointing, stability and launch loads. Extensive modeling and analyses have resulted in a (nearly) all Titanium design, based mainly on elastic deformation of components.

The main characteristics of the REM are:

- Volume of $\phi 280$ mm x 123 mm ($\phi 50$ mm central hole for optical beam passage).
- Mass of 6.8 kg.
- Eigenfrequencies of > 200 Hz (axial) and > 130 Hz lateral.
- Peak power consumption of 14 W (stepper motor).
- Life time requirements are 4000 cycles in 7 years in-orbit and maximum 12 years on-ground storage.

The present status is that the REM-EQM has been built, tested and qualified for MSG and that the production of two out of three flight models is nearly completed. The first launch is expected in October 2000. We intend to present the development process and the performance as tested. In the description of the design, attention will be given to the following components:

- The Guide module, based on six folded leafsprings, realized as monolithic components
 - A lever transmission, which connects to the Guide Module and reduces launch loads in this Module.
 - An SKF recirculating rollerscrew lubricated by ESTL (UK) using a combination of coatings.
 - A self lubricating planetary gear box (Vespel SP3 on Titanium).
 - A redundant winding stepper motor made by SAGEM.
 - A mechanical endstop for the stepper motor using a Vespel SP3 lever arm.
- Finally the performance results, focusing on stability, decenter and tilt, will be presented

A New Hybrid Air-spindle with Permanent Magnetic Bearing Incorporated for High Motion Accuracy and High Stiffness

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Introduction

Although the ultra-precision machining technology has been making a remarkable progress in recent years, the requirements of nanometers order form accuracy and subnanometers order surface roughness for X-ray optics still remains difficult to be met. One or two digits of the machining accuracy is expected to be improved. An air-spindle is usually used as the spindle of an ultra-precision machine, but its rotational motion accuracy can only attain around 30~50nm. To eliminate the rotational error of an air-spindle, an active air journal bearing system developed by Horikawa et al¹ introduced an air-bearing system to adjust the air gap between the rotor and the spindle using a set of air pads activated by piezoelectric transducer(PZT) actuators. Another research by Hara² proposed a hybrid air-spindle system combined with an electromagnetic bearing system.

In this paper, we proposed a new-type hybrid air-spindle combined with a permanent magnetic bearing system to decrease the rotational error and to improve the dynamic stiffness simultaneously.

Experimental apparatus

Configuration of proposed spindle

The configuration of the proposed new hybrid spindle is shown in Fig.1. Different from an ordinary electromagnetic bearing, four PZT actuators were axis-symmetrically arranged on the frame side of air-spindle, on each end of which a permanent magnetic block was mounted respectively. An attractive magnetic force which is theoretically inverse to the square of gap happens between each magnetic block and the laminated silicon steel plate ring assembled on the top of spindle axis. The gaps between the magnets and the silicon plate ring will be changed by controlling the elongation of PZT actuators so that the position of spindle axis might be adjusted precisely. To measure the spindle position and incorporate with the control system, a pair of capacitive sensors were mounted at 90° interval on the frame side. For the sensor target, an aluminum ring was fixed on the spindle and was diamond machined "on spindle" to decrease a possible eccentricity during assembling process.

The air-spindle used in this system was Canon AB-100RV with a built-in type brushless DC motor. Its radial stiffness was 52N/ μ m and axial stiffness was 304N/ μ m. The rotational accuracy was around 50nm. The PZT actuators applied were PFT-1070 type made by Chichibuonoda Co. The maximum elongation was 63 μ m under a maximum voltage of 140V and the band width was around 6kHz. The PZT actuator was preloaded to 900N, so it was proper to be used under attractive forces. The permanent

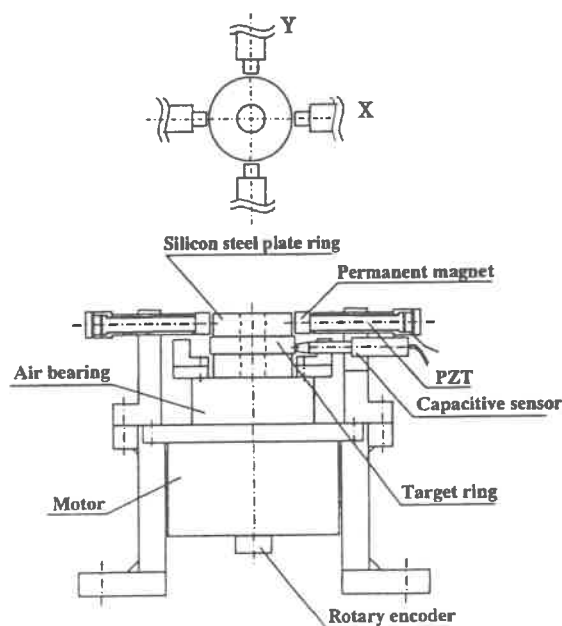


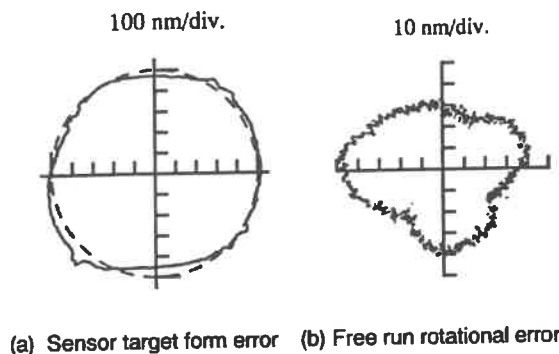
Fig.1 Configuration of proposed spindle

magnet blocks (NEOMAX, Sumitomo Special Metal Co.) had a maximum $B_s=1.2T$. The standard gap was kept 0.2mm while the PZT's working position is set to the middle of its elongation. The capacitive sensors were 3404HR-A2 (Japan ADE Co.) and its resolution was nominally 1nm.

Separation of form error of the sensor target ring

Since the sensor outputs include not only the rotational error but also the form error of the target ring, it is necessary to separate these two errors. To fulfill this task, the so-called Three Points Method³ was applied, three displacement sensor probes are used to detect the target roundness error and the spindle motion error simultaneously. Then the effect of motion error is cancelled in the differential output of probes and a deconvolving operation on the differential data yields the correct roundness error.

In this research, the angles between the three probes were chosen 106° and 137° considering the transfer function of the Three Points Method⁴ and a calculation was performed up to 150 undulations/rev. The measured roundness of the target ring is shown in Fig.2(a) and the rotational error under free run (no motor drive) is shown in Fig.2(b). A little variations happened among different measurements because of a temperature change. In most cases, the out of roundness was about 100nm and the rotational error was about 30~50nm. Therefore when we try to control the spindle rotational accuracy, it is clear that the form error could not be ignored and must be compensated during control.



(a) Sensor target form error (b) Free run rotational error
Fig.2 Calculation results of Three points Method

Construction of digital control system

A digital control system was constructed by using a floating point 32 bit digital signal processor DSP4300 (MTT Co.) with TMS320C31 on and 16 bit A/D and D/A converters and up/down counters. For the host computer, a personal computer (PC-9801BX4 NEC) was utilized. The pulse signals from the rotary encoder (1024ppr) were counted to measure the rotational angle and trigger the sample and control operations. Then the control inputs were sent to PZT via D/A converters and the PZT drivers where a reverse signal was fed to the opposed PZT. Finally, the necessary data were intermittently sent to the host computer to record the results.

Before designing the control algorithm, a series of experiments were performed to investigate the fundamental characteristics of the system while the spindle was not rotating. The maximum controlled displacement of spindle axis was around $\pm 100\text{nm}$ at a low frequency range. There found clearly a hysteresis curve in such a large displacement. But the system behaved almost linear for smaller amplitudes, which was confirmed by step response experiments. The resonant frequency was around 600Hz.

It is well known that the rotational error of an air-spindle is mainly caused by form errors of the elements and the disturbance from built-in drive motor, and therefore is highly repeatable within several nanometers. To eliminate these repeatable disturbances appearing in every revolution. In this paper, a repetitive controller⁵ is adopted. This is a kind of control method which uses the summation of rotational errors in former revolutions, so it can be considered as an integral controller for each angle of rotation.

The output of repetitive controller $v_i(\theta)$ at angle of rotation θ on i th revolution is given as following,

$$\begin{aligned} V_i(\theta) &= V_{i-1}(\theta) + k_r \cdot e_{i-1}(\theta + \omega) \\ &= \sum_{n=0}^{i-1} \{k_r \cdot e_n(\theta + \omega)\} \end{aligned}$$

Where,

$e_{i-1}(\theta + \omega)$: the rotational error at the angle of $(\theta + \omega)$ on $(i-1)$ th revolution

ω : phase lead

k_r : repetitive gain.

Fig. 3 shows the block diagram of control system. Because the digital signal includes high frequency noise due to the quantization, the system tends to diverge in a short time. To prevent this from happening, a FIR filter with an order of 40 and cut-off frequency of 300Hz was designed and inserted right after the repetitive operation. It was quite effective in stabilizing the system. On the other hand, since the repetitive controller saves all the error information by the last revolution, based upon which, the characteristics of the coming signal can be estimated, and the introduction of phase lead component ω is applied to compensate the system delay time and also improve the system stability.

Because the above mentioned algorithm can only eliminate the periodic disturbances which appear in some fixed frequencies, we still need to construct a parallel controller to compensate the real-time non-repeatable disturbances and improve the dynamic stiffness. In this study, a PI controller was applied.

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Experimental results

A series of experiments were conducted at different spindle speeds to investigate the characteristics of the proposed spindle and evaluate the motion accuracy and stiffness. An example of experimental results at a spindle speed of 900rpm is shown in Fig.4, where the dots express the locus of the spindle axis. The rotational accuracy was determined from the locus in a similar way of evaluating the out-of-roundness. The rotational accuracy of 1st revolution was 33nm. It was improved to 26nm by 20th revolution and after 40th revolution, it became stable and remained almost constant around 10nm. While

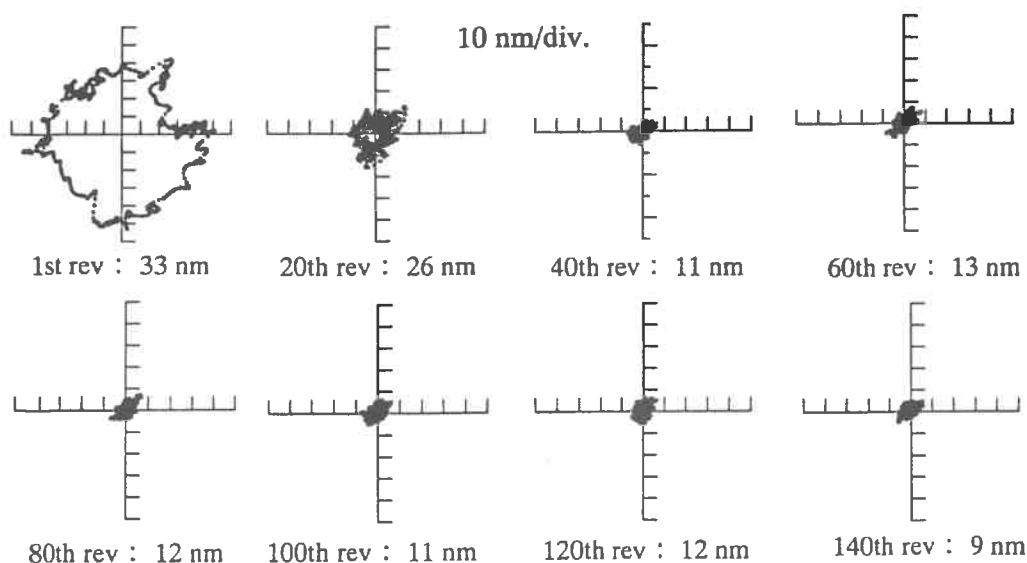


Fig.4 Loci of spindle axis at 900rpm when controlled

in rms value, it attained to 3nm. Using different control gains, similar experiments were repeated at spindle speed of 300, 600 and 1200rpm. Because of the disturbance from the drive motor, different rotational errors can be observed when uncontrolled. But similar results to Fig.4 were obtained at any spindle speed.

The stiffness of the proposed spindle was measured by exerting the compressed air to the silicon steel plate ring. The results are shown in Fig.5. It can be seen that the static stiffness improves from 52 N/ μ m to 83 N/ μ m only by assembling the permanent magnetic blocks even without control. While with control system performed, an infinite static stiffness was obtained up to 10 N, which was the maximum force the control system could compensate. In Fig.5(b), a comparison of compliance diagram with or without control is shown. The compliance with control is less by 5~10dB than that without control.

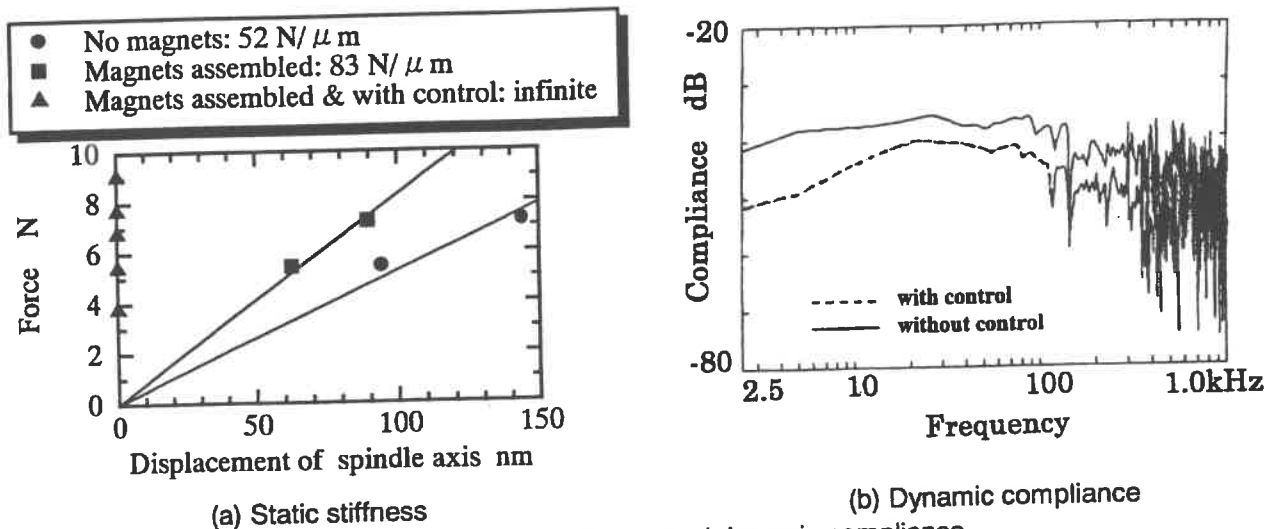


Fig.5 Static stiffness and dynamic compliance

Conclusion

In this paper, a new hybrid permanent magnetic bearing combined to an air-spindle has been proposed. It has been ascertained by experiments that the prototype spindle can obtain a high rotational accuracy around 10nm and in rms value around 3nm by using a repetitive controller. In the meantime, an infinite static stiffness and a much high dynamic stiffness with a 300Hz band width was also attained. Therefore a high precision machining accuracy can be expected using this proposed system as the spindle of a machining tool.

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On-machine Measurement of Angular Motion of a Spindle

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1. INTRODUCTION

Measurement of spindle error motions is necessary for ensuring machine tool performance. The error motion of a spindle includes the radial motion, the angular motion and the axial motion[1]. In these three kinds of error motions, angular motion is most difficult to measure accurately. Traditionally, the angular motion is calculated from the radial motions simultaneously measured at two spatially-separated positions along the axis of the spindle [1]. In this measurement, a precision test ball system that has two precision balls with their centers spaced some distance apart is generally used. To assure the accuracy of the measuring accuracy, the positions of the balls must be aligned with high precision to the axis of spindle rotation. The roundness of the balls is also needed to be less than the value of error expected for the spindle. However, these requirements are difficult to satisfy when extremely high measuring accuracy is necessary, such as the case of testing ultra-high precision machines. In addition, the measurement is also desirable to be performed under on-machine or in situ conditions in which the workpiece under machining is used as the sample instead of the precision test ball system.

The authors have proposed a simple error separation method, which is specifically suited to the angular motion measurement under on-machine conditions[2]. In this method, two angle probes, separated with a small angular distance, are fixed around a cylindrical workpiece mounted on the spindle. The probes are used to detect inclinations of axial generators of the workpiece. The angular motion and the local surface slope can be separated from each other and obtained from the output data of the two probes.

This method, which only uses two angle probes, is simple and effective enough for most cases. However, only the angular motion component in one fixed sensitive direction can be obtained, and errors may be introduced if the angular motion component in the direction perpendicular to the sensitive direction is large.

In this paper, we improve the proposed method by using three angle probes. This improved method can measure two dimensional components of angular motions accurately. The principle of the method is discussed in this paper. An optical sensor developed for this method, and some experimental results are also presented.

2. PRINCIPLE OF ERROR SEPARATION IN ANGULAR MOTION MEASUREMENT

Figure 1 shows the principle of the error separation method for angular motion measurement. A cylindrical workpiece is mounted on the spindle, and three angular probes with the same angular distance ϕ are fixed around the workpiece. The probes are used to detect inclinations of axial generators of the workpiece. The axes of the probes are set toward the center of the rotating axis of the spindle. The probes scan the same circumference of the workpiece while the workpiece is rotating.

If the outputs of the angular probes are $\tau_A(\theta_n)$, $\tau_B(\theta_n)$ and $\tau_C(\theta_n)$, they can be expressed as :

$$\tau_A(\theta_n) = \cos\phi E_{ax}(\theta_n) + \sin\phi E_{ay}(\theta_n) + r_z'(\theta_{n-1}), \quad (1)$$

$$\tau_B(\theta_n) = E_{ax}(\theta_n) + r_z'(\theta_n), \quad (2)$$

$$\tau_C(\theta_n) = \cos\phi E_{ax}(\theta_n) - \sin\phi E_{ay}(\theta_n) + r_z'(\theta_{n+1}), \quad (n=1, 2, \dots, N) \quad (3)$$

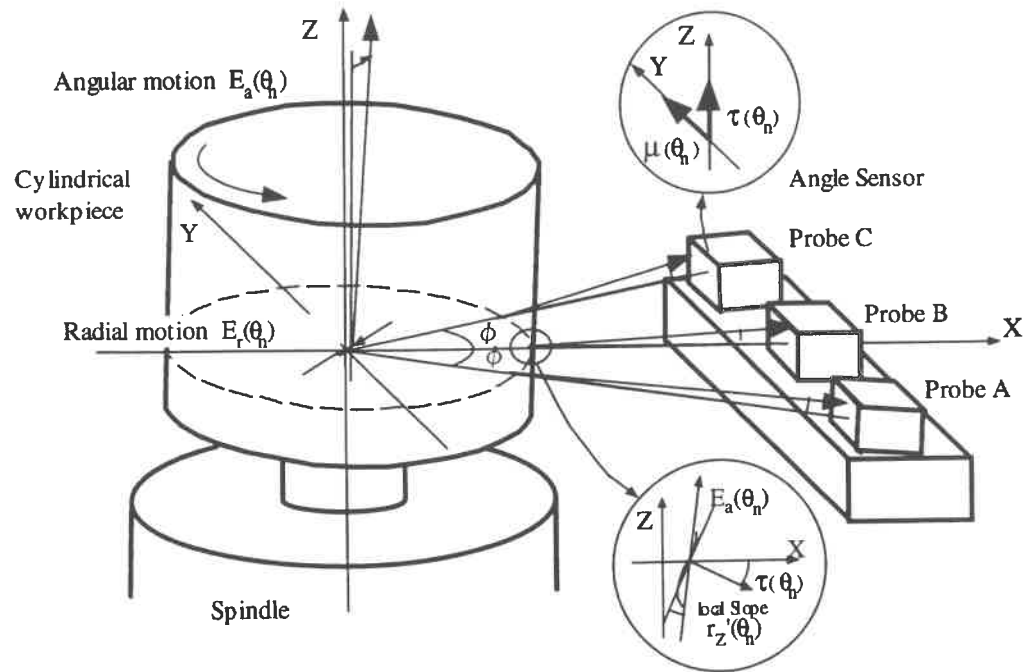


Figure 1 Principle of angular motion measurement

where, $r'_z(\theta_n)$ is the local surface slope of the workpiece along the axial generator, $E_{ax}(\theta_n)$ and $E_{ay}(\theta_n)$ are the angular motion components in the XZ plane and the YZ plane. θ_n is the rotation angle of the workpiece, and N is the sampling number in one revolution.

The differential output in which the angular motion term is canceled can be obtained as follows:

$$\begin{aligned} \tau_z(\theta_n) &= \tau_A(\theta_n) - 2\cos\phi \tau_B(\theta_n) + \tau_C(\theta_n) \\ &= r'_z(\theta_{n+1}) - 2\cos\phi r'_z(\theta_n) + r'_z(\theta_{n-1}) \end{aligned} \quad (4)$$

By using the Fourier transform technique, the local surface slope $r'_z(\theta_n)$ can be evaluated from Equation (4). The two dimensional components $E_{ax}(\theta_n)$ and $E_{ay}(\theta_n)$ of the angular motion can then be obtained from the probe outputs and the evaluated $r'_z(\theta_n)$.

Assume the angle probes are also sensitive in the XY plane, and their outputs are $\mu_A(\theta_n)$, $\mu_B(\theta_n)$ and $\mu_C(\theta_n)$, the two X-directional component $E_{rx}(\theta_n)$ and the Y-directional component $E_{ry}(\theta_n)$ of the radial motion of the spindle as well as the local surface slope $r'_z(\theta_n)$ of the circumference of the workpiece can be obtained by using the same technique described above.

3. EXPERIMENTAL SETUP

Figure 2 schematically represents the optical sensor developed to realize the proposed error separation method. A laser beam from a 10 mW laser diode is divided into three beams which are projected onto the workpiece. The reflected beams from the workpiece are received by three two-dimensional angular probe units. The angular probe unit, which consists of a lens and a photo detector, utilizes the principle of autocollimation. The photo detector is placed near the focal point of the lens. The focal distance of the lens is 30 mm. 4-cell photo diodes are used as the photo detectors so that the angular probe units can be sensitive

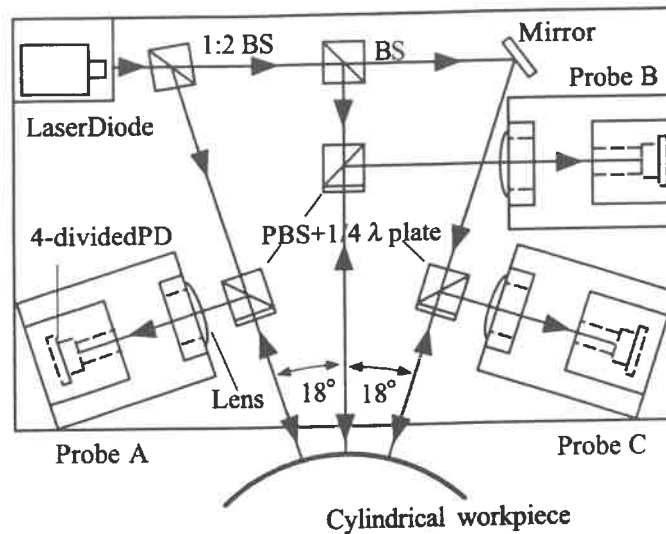


Figure 2 Schematic view of the optical sensor

in two dimensions.

4. EXPERIMENTAL RESULTS

The sensor was used to measure the error motion of an air spindle. A cylindrical workpiece with a diameter of 80 mm was mounted on the spindle and was used as the sample. The outputs of the probes were sampled by a multiple channel 12 bit A/D converter. The rotation angle of the spindle at each sampling point was measured by an optical encoder.

Figure 3 shows the results of stability test of the sensor. The two-dimensional outputs of each probe are shown in this figure. The test term was 300 seconds. The probes showed a stability of better than 0.1 arcsec.

Figures 4 to 8 show the results of error motion measurements. The two dimensional outputs of probe B in one revolution are shown in Figure 4. The sampling number was 1000. Figure 5 shows the measurement results of local slopes. Figures 6 and 7 show the results of error motions. Figure 6 shows the two-dimensional components of the angular motion, and Figure 7 shows those of the radial motion. The

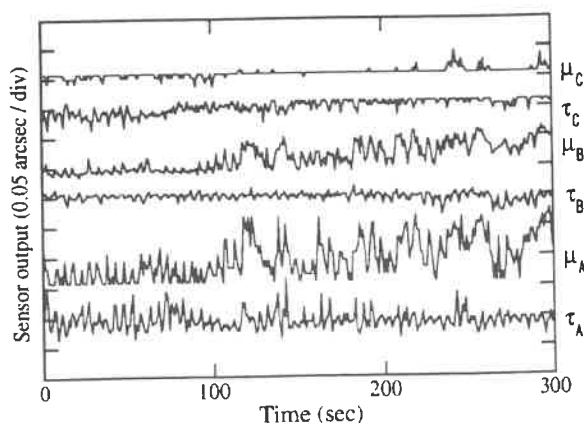


Figure 3 Results of stability test

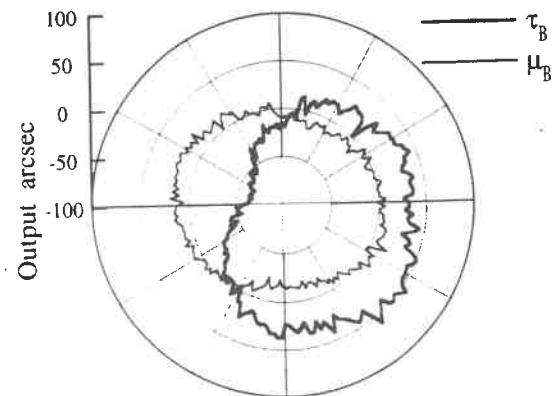


Figure 4 Two dimensional outputs of probe B

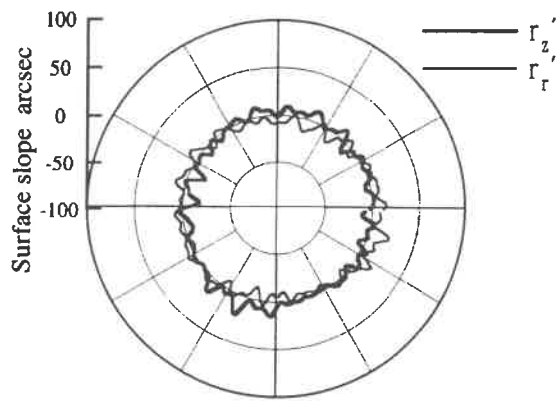


Figure 5 Measurement results of local slopes

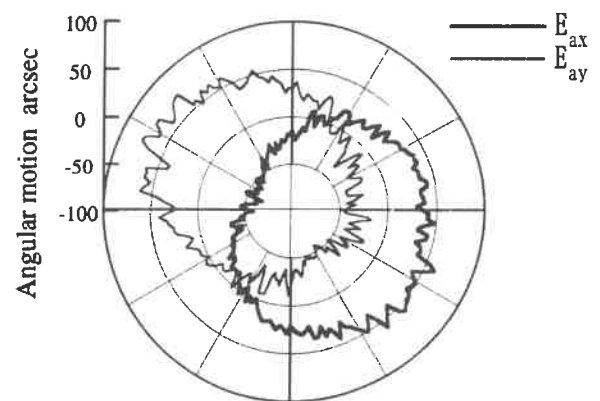


Figure 6 Measurement results of the angular motion

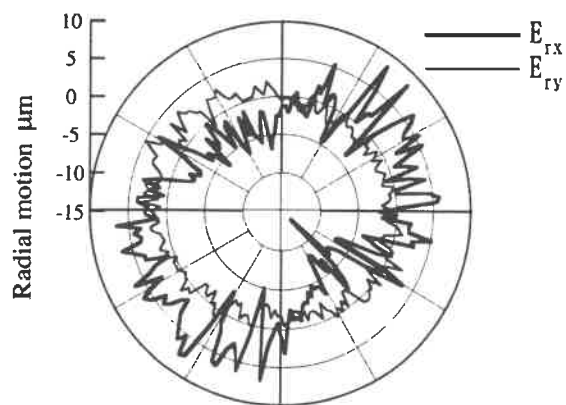


Figure 7 Measurement results of the radial motion

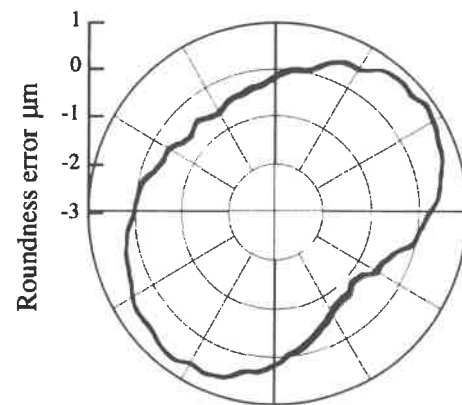


Figure 8 Measurement results of the roundness error

maximum angular motion and radial motion shown in these figures were approximately 120 arcsec and 13 μm , respectively. A large part of the error motions are considered to be introduced from the coupler connecting the encoder and the spindle in the experimental spindle system. Figure 8 shows the roundness error obtained by integrating the local slope. The roundness error was approximately 2 μm .

5. CONCLUSIONS

An optical system consisting of three optical angle probes with high resolution and stability has been developed to realize a new error separation method for angular motion measurement. The radial motion can also be obtained at the same time. Some experiments have been performed to confirm the effectiveness of the proposed system.

ACKNOWLEDGMENT

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Three-Degree-of-Freedom Optic Mount for Extreme Ultraviolet Lithography

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Abstract

A mechanism to finely align optics for extreme ultraviolet lithography applications is presented. The mechanism is a small motion parallel link manipulator with flexure joints that exhibits nanometer level positioning capability. Performance results of a prototype system are given in this paper.

Keywords: precision stage, flexure, kinematic design, parallel link manipulator, error budget, thermal stability, EUVL

Introduction

The projection optics required for extreme ultraviolet lithography (EUVL) must be precisely figured to sub-nanometer rms surface figure accuracy over 50 to 150 mm apertures and multi-layer coated for high reflectivity. This requires extending the state of the art in optics manufacturing, multi-layer coatings and optics metrology. Accordingly, the opto-mechanical system design must represent a small proportion of the total figure error budget. In addition, final alignment of optics in the projection optics system requires motorized adjustments with nanometer-level resolution, sub-millimeter range, and a high degree of dimensional stability. A prototype actuation system meeting these requirements has been built.

The information required to calculate the desired mirror positions comes from interferometric measurements of the assembled optical system's wavefront over multiple field points. The typical optic requires motorized adjustments in three degrees of freedom, namely, translation along the optical axis (piston) and rotation about the two axes in the plane of the optic (tip and tilt). Once aligned, the actuation system is disabled until realignment is required at some future date. The actuation system is not presently intended to provide real-time imaging system compensation during EUV lithography operation. This paper presents our design solution for the actuated three-degree-of-freedom optic mount. The discussion includes system design issues, error budget analyses related to satisfying long-term and short-term dimensional stability, and experimental results from a prototype system.

System Design

System design requirements for the actuation system flow down from the optical design and stability

requirements for a scanning lithographic machine. The resolution requirement for actuation is less than 10 nm in translation and less than 30 nrad in rotation. Furthermore, the system must demonstrate long-term dimensional stability, high dynamic stiffness, and have little effect on the optic's surface figure when actuated to its limits of motion (± 100 microns).

A parallel mechanism allowing translation at three points around the periphery of the optic cell is chosen to meet these requirements. Monolithic flexures provide friction-free joint movement and superior temporal dimensional stability. This particular actuation system uses three identical flexure mechanisms that transmit radial motion from high-resolution piezo screws to a 5:1 reduced axial motion at the cell's periphery. Simultaneous and equal motion of the three piezo screws provides pure axial translation of the cell (piston) while differential motion of the motors provides rotation of the optic cell (tip and tilt).

The piezo screws have a resolution generally less than 30 nm and with the 5:1 transmission ratio provide the required resolution. Capacitance sensors located on the optic cell near the three-actuation positions provide feedback for the optic cell movements. Capacitance sensors have subnanometer resolution with low thermal and electrical drift characteristics. The system without the capacitance gauges is shown in Figure 1.

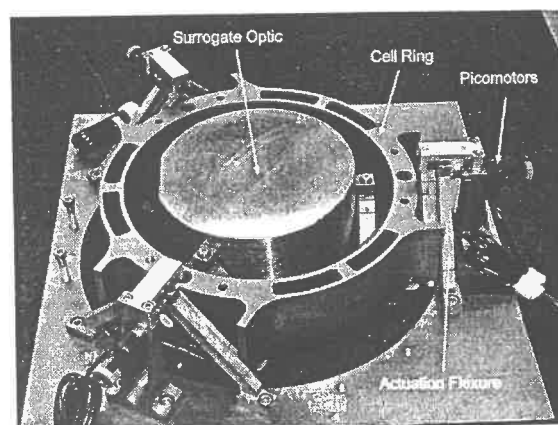


Figure 1: Prototype Actuation System

The optic cell that holds the optic is actuated in three degrees of freedom by actuation flexures that are located 120 degrees apart around its perimeter. Each actuation flexure is electrodischarge machined (EDM) from a single piece of Super Invar to maximize dimensional stability. The monolithic flexure has two

functional parts, described as the transmission flexure and the constraint flexure, as shown in Figure 2.

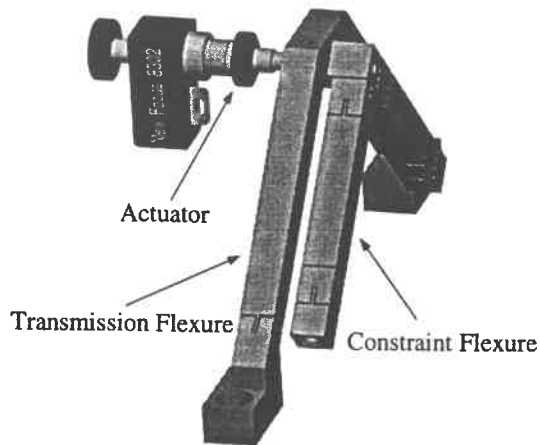


Figure 2: Detail of a single actuation flexure

The side of the actuation flexure that attaches to the projection optics housing is called the transmission flexure. The blades on this flexure form a hinge axis near their base so that movement of the piezo screw causes rotation of the transmission flexure about this axis. The role of the constraint flexure is to kinematically locate the optic cell with respect to the transmission flexure in two degrees of freedom. The plane of the blades in the transmission flexure intersects the instantaneous center of rotation of the constraint flexure. This condition maximizes the stiffness in the two constraint directions. Since the constraint flexure is offset from the transmission flexure, rotation of the transmission flexure produces displacement of the cell in the plane of the constraint flexure. Motion at the piezo screw is accommodated by compliance in the constraint flexure. This compliance also serves to isolate the optic cell from actuation related loads that could cause changes in the optical substrate figure.

Error Budget

An error budget provides a mechanism with which to evaluate tradeoffs in design and to predict overall system performance. The EUVL projection optics system is dominated by two basic sets of performance requirements. The first requirement is preservation of surface figure of the optics and the second is stability of the dimensional relationship between the optic surfaces. The total surface figure error budget is less than 0.25 nm rms over the clear apertures of the optics. Surface figure error is defined as the figure error between spatial frequency limits from the clear aperture to one inverse millimeter. Of the many contributors to the figure error budget, the actuation flexure contribution is set at less than 0.08 nm rms. Essentially, the actuation system can not influence the figure of the optics.

The dimensional stability requirement is broken down into scanning stability and long-term stability. The scanning requirement corresponds to a time scale related to that required for a single image scan. For the prototype EUVL system this is limited to approximately 10 seconds. Vibration is the primary error source related to this requirement. The long-term requirement is derived from the desire for an aligned imaging system to stay aligned within the operational environment for long periods of time. Thermal effects during operation and temporal dimensional changes of systems are the main issues affecting long term stability. A time scale ranging from many months to a year is chosen for this error budget. Effectively, the time scale sets the period after which major imaging system realignments must be performed. The dimensional stability requirements for each optical element for scanning and long term are given in Table 1. Long term stability of the joints, materials, and piezo screw in the actuation system are the topics of ongoing tests.

DOF	Scanning Error (rms)	Long Term Error
θ_x	3 nrad	$\pm 0.50 \mu\text{rad}$
θ_y	3 nrad	$\pm 0.50 \mu\text{rad}$
X	1 nm	$\pm 0.29 \mu\text{m}$
Y	1 nm	$\pm 0.29 \mu\text{m}$
Z	4 nm	$\pm 1.00 \mu\text{m}$

Table 1: Positioning Error Budget

Modal Analysis

A finite element modal analysis and experimental modal testing have been performed on the prototype actuation system. The finite element model (FEM) was constructed using shell and solid elements and the model was constrained similarly to the actual imaging system. The finite element analysis predicts the first mode of vibration to occur at 133 Hz. This is a longitudinal mode of vibration that is dominated by compliance in the actuation flexures.

The results of the experimental modal testing on the prototype actuation system agreed with the finite element model. The first dynamic mode occurred near 133 Hz as predicted by the FEM. The magnitude of the experimental frequency response function is shown in Figure 3. The range of the frequencies shown in Figure 3 is from 0 to 800 Hz.

A preliminary dynamic analysis using a low order representation of the EUVL projection optics system and the isolation systems has been completed. The results suggest that all structural modes of vibration should be greater than 150 Hz to ensure meeting the scanning stability goals. Based on these results, design changes have been implemented in the actuation system

that increase stiffness and raise the natural frequency to near 200 Hz. This was accomplished by adding structural reinforcement between the legs of the transmission flexures and increasing the blade width at the base of the flexures. Furthermore, increasing the blade thickness on the blades closest to the instant center increased the axial stiffness of the constraint flexures.

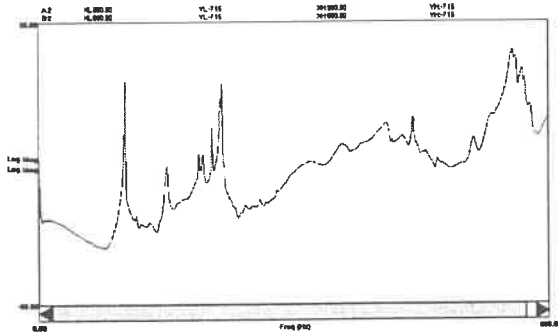


Figure 3: Frequency Response Function

Kinematics & Controls

The kinematics of this system can be modeled similar to traditional parallel link mechanisms such as the Stewart Platform Manipulator. For small amounts of z motion, the flexures can be kinematically modeled in terms of spherical, revolute and prismatic joints as shown in Figure 4.

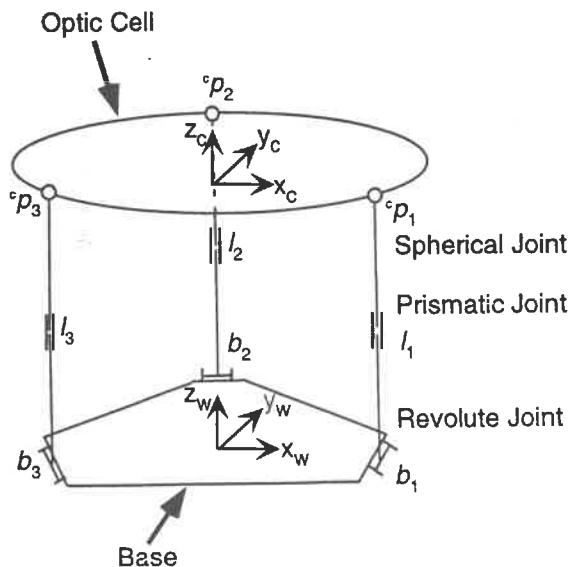


Figure 4: Kinematic Links and Parameters Of the Actuation System

Mobility analysis shows that the system has three degrees of freedom as expected:

$$M = 6(m - j - 1) + \sum DOF = 6(8 - 9 - 1) + 15 = 3$$

where m is the number of members (8), j is the number of joints (9) and DOF is the total degrees of freedom of

the joints ($15 = 3 \cdot 3 + 3 \cdot 1 + 3 \cdot 1$). The chosen degrees of freedom are rotations about x and y , and displacement in z , all occurring about the local coordinate system as defined by the chief ray point (point C) of the optic. This coordinate system is shown in Figure 4. The spatial coordinates of the base points (b_1, b_2, b_3) are defined in the world coordinate system (W), and the spatial coordinates of the platform points (p_1, p_2, p_3) are defined in the chief ray coordinate system (C).

The length of the prismatic joints can be written as:

$$l_1 = \sqrt{(T \cdot p_1 + d - b_1)^T (T \cdot p_1 + d - b_1)}$$

$$l_2 = \sqrt{(T \cdot p_2 + d - b_2)^T (T \cdot p_2 + d - b_2)}$$

$$l_3 = \sqrt{(T \cdot p_3 + d - b_3)^T (T \cdot p_3 + d - b_3)}$$

where T is the RPY (roll, pitch, and yaw) rotation matrix about z_c , y_c , and x_c , and d is the location of the C coordinate system relative to the W coordinate system. Note that the rotation about z (roll) is zero.

$$T = \text{RPY}(0, \theta, \psi) = \begin{bmatrix} \cos \theta & \sin \theta \sin \psi & \sin \theta \cos \psi \\ 0 & \cos \psi & -\sin \psi \\ -\sin \theta & \cos \theta \sin \psi & \cos \theta \end{bmatrix}$$

$$d = [0 \quad 0 \quad z]^T$$

These equations are used to direct the piezo screw displacement within a low bandwidth control scheme based on the capacitance gauge feedback. The graphical user interface of the controller receives user input as tip/tilt/piston motion about the chief ray point and computes the required movements in the joint space (l_1, l_2, l_3) using the kinematic equations. The control system drives the motors until the capacitance gauges reach their desired values.

Surface Distortion from Actuation System

To estimate the surface figure deformations due to actuation inputs from the piezo screws, detailed finite element models were created and analyzed using structural finite element model software. The optic is supported within the cell by three bipod mount flexures, similar to the previously mentioned constraint flexures. The collection of the three bipod mount flexures constrain the rigid body motion of the optical substrate, while partially isolating the substrate from disturbance forces and moments that deform the surface figure. The analyses considered the anticipated full range of motion of the optic, ± 100 microns in piston. Surface mesh plots of the clear aperture were generated using the extracted raw surface displacement data. For a 100 micron z displacement, the rms surface deformation is estimated to be 0.05 nm. Figure 5 shows an example of surface

deformations within the clear aperture boundary for an actuated optic with a nominal diameter and thickness of 170 mm and 65 mm respectively (clear aperture is 120 mm in diameter).

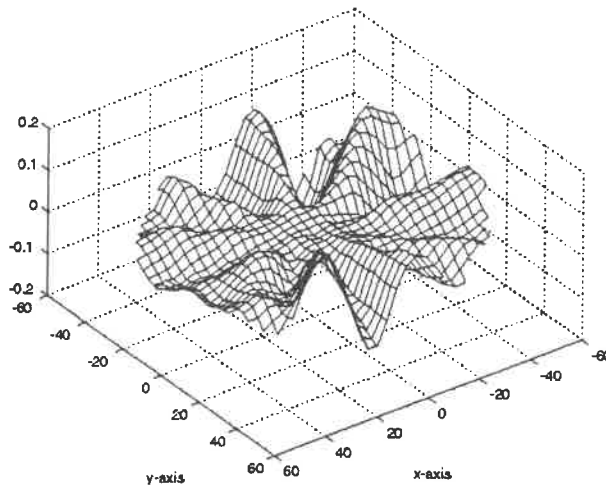


Figure 6: Mesh plot of surface deformations within the clear aperture (x,y: mm scale, z: nm scale)

Actuation Prototype Testing Results

The positioning capability of the actuation system was tested in a $\pm 0.5^\circ\text{C}$ temperature controlled environment. The positioning results for commanded Z motion are shown in Figure 7. The solid line shown in the figure represents the commanded motion and the dotted line is the actual position (capacitance gauge reading) of the actuated optic. The graph shows movement as small as 2 nm and as large as 50 nm. In simpler bench top testing, the system appears to demonstrate excellent dimensional stability, but the extreme demands of the EUVL imaging system require very exacting long-term measurements of dimensional stability. These tests are presently being initiated. Preliminary testing shows nanometer level stability for 0.5 minutes and longer in some cases. Ten-minute drift of all the capacitance gauges is shown in Figure 8. Thermal stability of 5 nm or less has been seen during this period.

To verify the commanded position, a displacement and angular measuring interferometer (He-Ne class II laser) is used to measure the commanded motion at the chief ray point. As configured, the interferometer has a 10 nm distance resolution and a $0.5 \mu\text{rad}$ angular resolution.

The effect of dead-air-path is reduced during the measurements by minimizing the distance between the mirrors and the interferometer. A small sample of the results from the testing of the positioning and angular displacements is given in Table 2. Motion about the unactuated axes is also measured during movement of controlled axes and the results are reported in Table 3.

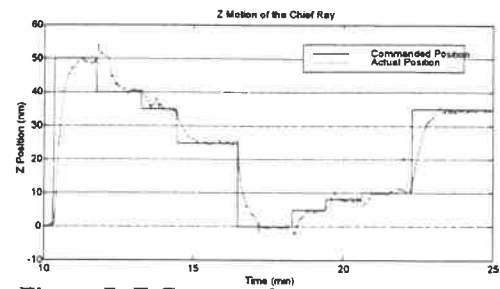


Figure 7: Z Commanded/Actual Motion Of the Chief Ray Point

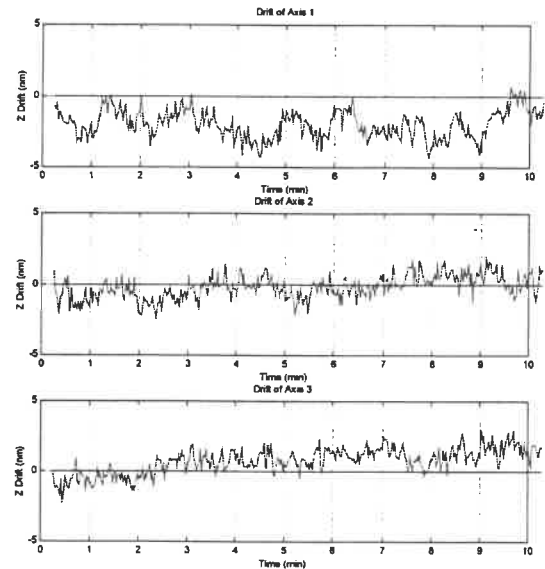


Figure 8: Short Term Capacitance Gauge Drift

Trial	Dir.	Commanded Position	Measured Position
1	Z	100 nm	$100 \pm 10^\dagger$ nm
2	θ_x	$5 \mu\text{rad}$	$4.84 \pm 0.484^\ddagger$ μrad
3	θ_y	$28 \mu\text{rad}$	$27.6 \pm 0.484^\ddagger$ μrad

† Resolution of the DMI

‡ Resolution of the AMI

Table 2: DMI/AMI Measurements

Trial	Commanded Axis	Measured Axis
1	Z=-100 nm	X<10 † nm
2	$\theta_x=5 \mu\text{rad}$	X<10 † nm
3	$\theta_y=5 \mu\text{rad}$	X<10 † nm

† Resolution of the DMI

Table 3: Undesirable Motion Of the Actuation Cell

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The High Precision Linear Motion Table With a Novel Rare Earth Permanent Magnet Biased Magnetic Bearing Suspension

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Abstract

An electromagnetic bearing needs large amount of bias current to support its own weight so that its load capacity is limited and problems caused by heat generation can be arose. In order to remove these drawbacks of the electromagnetic bearing, we developed a novel permanent magnet biased electromagnetic bearing using a rare earth permanent magnet. The permanent magnet provides a bias flux and generates a bias force. The electromagnet can increase or reduce a flux of the permanent magnet and the flux path of the electromagnet does not pass through the permanent magnet that the permanent magnet will not be demagnetized. One actuator can support one control axis and it is composed of one electromagnet and two permanent magnets that it needs one current amplifier for one control axis. We adopt this newly designed electromagnetic bearing system to the 6 D.O.F. linear motion table as a magnetic suspension^[1,2,3]. The linear positioning of the table with a 100 mm stroke is driven by brushless DC Linear motors and the other 5 D.O.F. of the table are controlled by the digital controller with magnetic bearing actuators. The novel electromagnetic bearing system guarantees the high load capacity and it needs less current than conventional type of the electromagnetic bearing so that it gives low power consumption and elimination of problems caused by the heat generation.

Keywords: novel magnetic bearing, rare earth permanent magnet, low power consumption, high load capacity, linear motion table

1. Introduction

The development of the permanent magnet with the rare earth material such as NdFeB or SmCo increase the maximum BH product and the maximum coercive force that it can be applied to many of the industrial machinery such as electric motors, solenoid valves and magnetic actuators. But the permanent magnet is a passive element that it is originally unstable and it is impossible to levitate the permanent magnetic levitation system alone. Therefore the permanent magnetic levitation system is applied to the suspension system with additional levitation or support apparatus. Occasionally, the turbomolecular pump system uses the passive magnetic bearing to support the radial direction but the thrust bearing is an active one. Recent days, the permanent magnet is applied to reduce the bias force of the electromagnets and therefore can reduce the heat generation from the electromagnets. Sortore^[4] made the radial magnetic bearing with the permanent magnet bias. In

this system, the flux path of the permanent magnet is partially superposed with that of the electromagnets and he successfully reduced the power consumption and heat generation. Boden^[5] made the commercial product of the magnetic bearing using the permanent magnets.

In the linear X-Y table with the active magnetic levitation system, the additional load of the upper platen is added to the lower levitation system that the bias current should be increased. But the maximum load carrying capacity of the active electromagnetic bearing is limited by bias current that a size of the table system should be increased exponentially with the increase of the load carrying capacity of the active magnetic bearing. Furthermore, in order to support the weight of the table system, more current is needed and therefore more heat is generated. The excessive heat can make the serious problems to the whole linear X-Y motion table system. In order to increase the maximum load carrying capacity of the active magnetic bearing and to reduce the current for the

electromagnets, it needs to adopt the permanent magnet for the generation of the bias force. The purpose of this paper is the development of the permanent magnet biased active magnetic bearing system. There are two design goals. One is to develop the active magnetic bearing system with high load carrying capacity using the permanent magnet bias and the other is to design a small size of the magnetic actuator applying to the X-Y linear motion table system.

2. Development of the Actuators

There are many examples using the permanent magnet to eliminate the gravitational force. But almost of them are rotational or stationary configurations that it is hard to apply them into the linear motion system directly. In the development of the permanent magnet biased active magnetic actuators for the linear motion system, three design rules are adopted.

Design rule 1

The flux paths of the permanent magnet may partially or fully superposed with that of the electromagnets. To reduce the demagnetization effect of the permanent magnet, the flux paths should be designed carefully that the paths of the electromagnets must not pass through the permanent magnet.

Design rule 2

The permanent magnet is a material with a hysteresis effect so that the flux of the permanent magnet should be applied to generate the bias force only, not the perturbation force. Therefore, the location of the permanent magnets should be eluded from the pole face area.

Design rule 3

The active magnetic bearing should have some bias flux and it may be increased or reduced by adjusting the current amplifier. However, the permanent magnet is always saturated and it may saturate the flux paths of the electromagnets which are made of iron. Therefore, the flux density of the whole loops in the operating point should be maintained below half of the saturation limit.

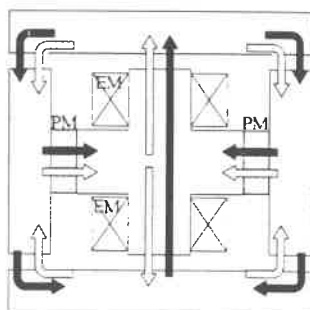


Fig. 1 Simplified Model of Magnetic Actuator
Black arrows: flux for electromagnet
Gray arrows: flux for permanent magnet

According to the three design rules, the new type of the permanent magnet biased magnetic actuator is designed and it is shown schematically in Fig. 1. The new model is composed of stationary and moving parts and the moving parts are divided into four parts. The stationary parts are the upper and the lower rectangular. In the moving parts, the iron poles of both sides make the return flux paths and the cross shape of iron pole make the main flux path. These two electromagnets are located in the upper and lower legs of the iron cross and the two permanent magnets are located between left and right legs of the cross shaped part and the iron poles for the return flux path. The north poles of the permanent magnets face to the iron cross and the two electromagnets are connected in series that the polarities of the two electromagnets are the same. The electromagnets are connected in series that it needs only one current amplifier for the control of one control axis. The flux of the permanent magnets generate the bias force to the upper and lower stationary parts. Nevertheless, the net flux which passes by the permanent magnets is always the same and goes to the same direction of the permanent magnets that it does not need to worry about the demagnetization effects or the hysteresis effects.

The equivalent circuit for the simplified model given in Fig. 1 is shown in Fig. 2. The equation (1) is governing equation of the equivalent circuit and it is composed of flux from the electromagnet and from the permanent magnet. The two terms are linearly superposed.

$$\begin{bmatrix} 2 \frac{g_0 - x}{A} + \frac{t_m}{\mu_r A_m} & -\frac{t_m}{\mu_r A_m} \\ -\frac{t_m}{\mu_r A_m} & 2 \frac{g_0 + x}{A} + \frac{t_m}{\mu_r A_m} \end{bmatrix} \begin{bmatrix} \phi_1 \\ \phi_2 \end{bmatrix} = \begin{bmatrix} \mu_0 \cdot N \cdot i + \frac{B_r t_m}{\mu_r} \\ \mu_0 \cdot N \cdot i - \frac{B_r t_m}{\mu_r} \end{bmatrix} \quad (1)$$

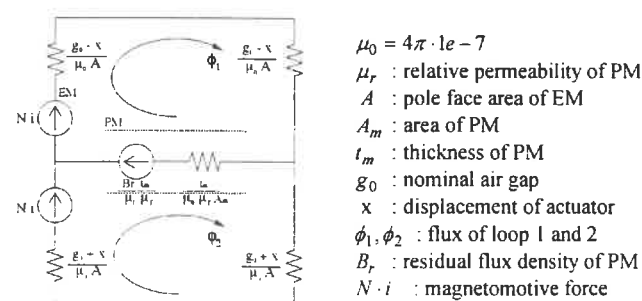
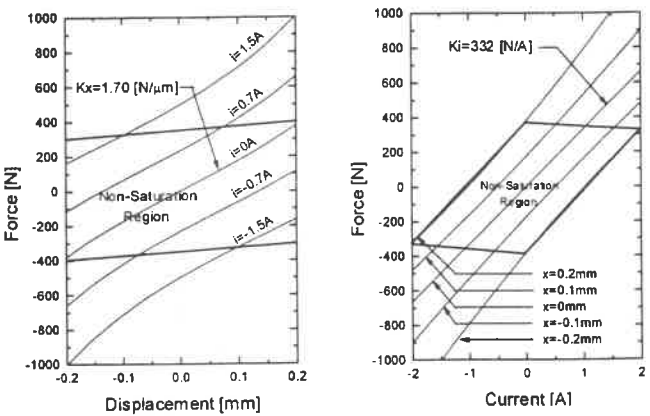


Fig. 2 Equivalent Circuit of the Simplified Model

The analysis results of Fig. 3 show the relationship between the displacement and attraction force, and between the current and attraction force are almost linear in whole operating range. As a result, $k_x = 1.7 \text{ [N/}\mu\text{m]}$ and $k_i = 332 \text{ [N/A]}$ at the nominal operating point and operating range is limited by the saturation of the air gap at the pole face. Fig. 4 shows the flux lines of the simplified model.

The flux lines of the permanent magnets are symmetrically distributed at the center position that they only give the bias force. The flux lines of the electromagnets pass through the center part of cross element and return to the flux ring that the flux of electromagnets cannot levitate the system alone. The superposed result of the two flux lines shows that the two flux are added at one side and subtracted at the other side. The differences of flux between the two sides give the levitation force.



(a) current vs. force (b) displacement vs. force
Fig. 3 Analysis Result for the Equivalent Circuit

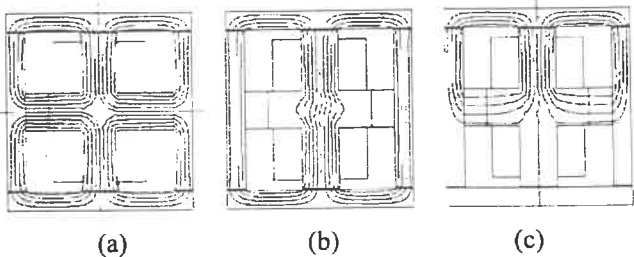


Fig. 4 The Flux Lines for the Simplified Model
 (a) flux lines of the permanent magnet
 (b) flux lines of the electromagnet
 (c) linearly superposed result

3. The Experiment of Single Degree of Freedom

In the analysis results from the simplified model, it is impossible to find out the correction factors due to the leakage effect, eddy current effect, iron loss and hysteresis effect. So, it needs to test the single degree of freedom model to find out the exact performance of the suggested model. The test rig is given in Fig. 5 and the system schematics are given in Fig. 6. The vertical motion of the actuator is monitored by the eddy current type proximitors, which is located at the upper stationary plate. The plant is composed of test rig, sensor system, A/D and D/A converter, servo amplifier, power supply, personal computer and DSP board for the PID control algorithm. The transfer function for the system is given in (2). In this equation, the

characteristics of the system can be explained by two parameters k_x and k_i . They can be derived from the governing equation of (1) but they have discrepancy to the measured results because the simplified model did not consider the various types of losses. In order to find out the correction factors between the analysis and experiment, the k_x and k_i are measured as following schemes. The relationships between the displacement, current and external load are measured after the system is controlled by PID algorithm which make the steady state error into zero.

$$G(s) = \frac{k_i}{ms^2 - k_x} \quad (2)$$

The measured result shows that the system can be regarded as a linear system and the two graphs can be expressed as equation (3) and (4) respectively.

$$F = a \cdot i + b \quad (3)$$

$$i = c \cdot x + d \quad (4)$$

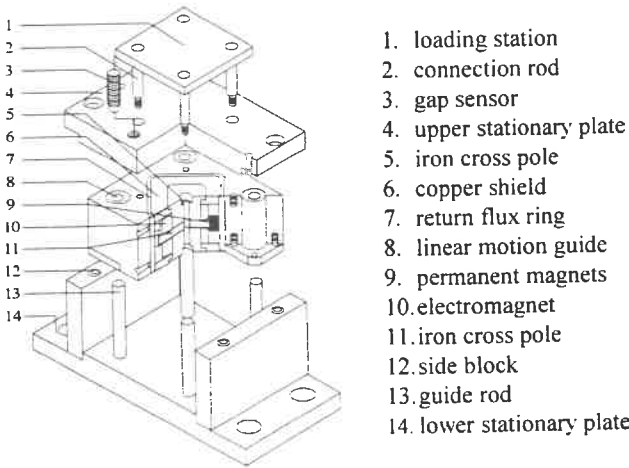


Fig. 5 The Test Rig for One Degrees of Freedom

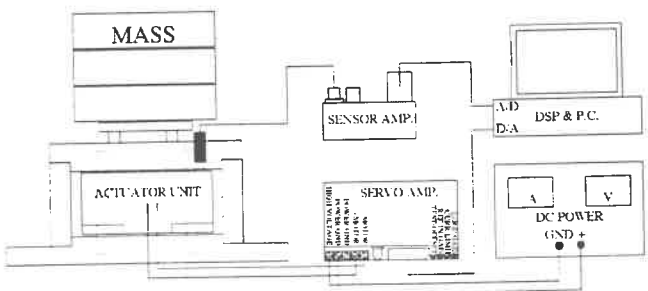


Fig. 6 System configuration for the test rig

From the Fig. 7 (a), the slope of the graph is a measured current stiffness and therefore the constant a of equation (3) is k_i . The position stiffness cannot be derived directly from the Fig. 7 (b). Substituting the current i at the equation (4) into equation (3), the position stiffness can be obtained. The discrepancy between the experimental and the analytical result shows that some correction factors should be included

to correct the two results. One of the correction factors is originated from the permanent magnets and the other is from the electromagnets. We assume that the discrepancy is due to the error of the flux density B by various loss effects. Therefore these two correction factors are obtained from the measured and the analyzed results that the correction factor for permanent magnets is 0.7 and that for the electromagnets is 0.9. The experimental result for $k_r = 1.1$ [N/ μ m] and $k_f = 220$ [N/A]. To find out the maximum load capacity of this actuator, the initial levitation test is performed by increasing the external load and it can support maximum 230 N. The size of the actuator is 40 x 40 x 35mm and magnetomotive force at this situation is 350 A.turns ($i=1$ A).

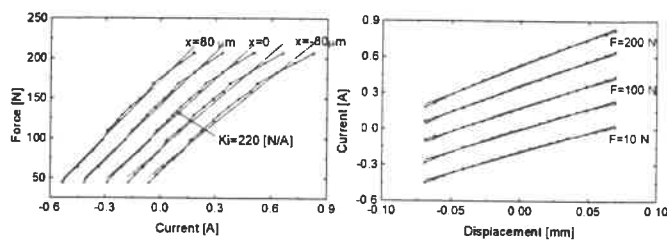


Fig. 7 Experimental Results for the single degree of freedom system

- (a) current vs. force relationship for various reference position
- (b) displacement vs. current relationship for various external load

4. The Linear X-Y Motion System

The high precision linear X-Y motion table can be used in many industrial applications and this table system is developed for the purpose of the inspection stage for the semiconductor wafer and the stage for the high precision electric discharging machine. The assembled shapes of the X-Y table system are given in Fig. 8. The linear motion system is composed of three parts. The moving stage at the top is the table for Y direction and the guide rail for Y direction is connected to the moving stage for X direction. The guide rail for X direction is connected to the base plate. Each table has four actuator modules and all the modules have the same shape. Each module has two actuators. One is the permanent magnet biased active magnetic actuator for vertical levitation and the other is for the horizontal levitation. The pair of the actuators for horizontal levitation at the same longitudinal direction make one axis control plant. Therefore, the four actuator modules compose five degrees of freedom motion control plant and the remain axis is controlled by the linear motor. The capacitance type displacement sensors^[3] are located just near by the each

actuators and in order to reduce the noise from the various circumstances, the amplifiers for the sensor is attached in the table body so that the finally conditioned analog signals are transferred to the controller system.

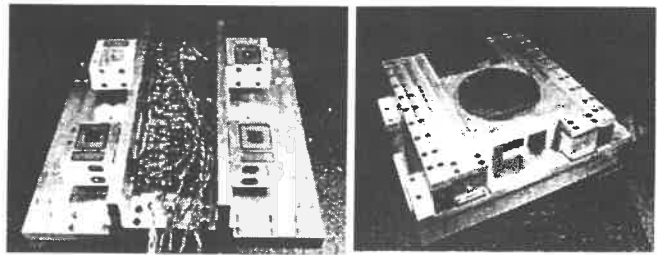


Fig. 8 Assembled shapes of the X-Y Table System

5. Conclusion

We have developed a new type of the permanent magnet biased magnetic actuator and adapted to the linear X-Y motion table system. A single degree of freedom test rig was constructed and tested for the basic research of the extended system. We have measured load capacity and bearing characteristics. The observed performance is reasonably matched with the theoretical prediction. An improved load capacity of the actuator is suitable enough to adapt to the heavy load carrying system, so linear X-Y motion table system was manufactured and is being tested.

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Development of a Precise Positioning Stage Traveling on Micro-pitch Racks

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1. Introduction

Recently, a precise positioning stage has been installed in a precise machine tool, a measuring instrument and a semiconductor manufacturing equipment. These stages require not only mechanical performance with high stiffness, accurate guidance and the minimum lost motion but also control performance with precise positioning, minute resolution and dependable repeatability of positioning and smooth driving. Some integrated precise stages with a devised drive mechanism and an intelligent feedforward and feedback control system have been developed.¹⁾²⁾³⁾ One integrated stage consists of a ball screw, an air slider and a laser interferometer.¹⁾ Some stage is formed into a linear motor with a precise linear encoder combining a grating and a laser.²⁾ Other stage is composed an inchworm mechanism by using piezoelectric actuators.³⁾ However, the first stage is bulky because individual instruments are assembled without merging. If stiffness of the ball screw and the air slider should be more rigid and measurement accuracy should be higher, those elements become bigger and heavier. Control system is complicated for combining those elements. Although the second stage is compact, a control system is complicated because the linear motor cannot control the position without regulating of electric current accurately. Control system of the third stage is constructed easily because inchworm motion can be controlled with an open-loop system, yet the stage can not position every step without slipping even if driving plates grip on a guidance rail tightly.

In this paper, we present a novel precise positioning stage traveling on micro-pitch racks with which eight micro-pitch teeth mesh alternately. The stage can position every $60\text{ }\mu\text{m}$ over the travel distance of 40 mm in a procedure of the inchworm motion. The structure is compact and the open-loop control system is used, besides the stage does not slip in the inchworm motion.

2. Structure of the precise positioning stage

The precise positioning stage is consists of a stage board and an elastic hinge frame as shown in Fig. 1 and Fig. 2. The stage board is mounted on two parallel linear guides laid on the base board, and the stage board traverses right and left accurately. The elastic hinge frame is inserted between the base board and the stage board, and the left end of the elastic hinge frame is fixed on the base board.

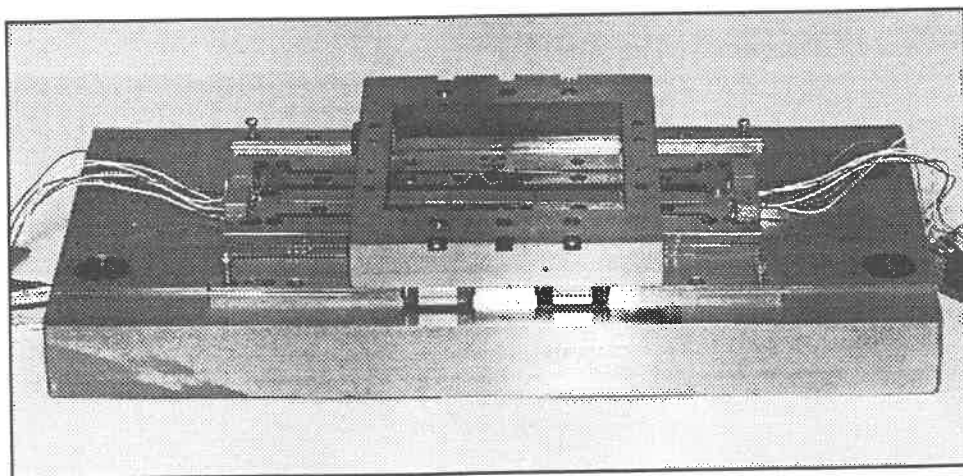


Fig. 1 The precise positioning stage traveling on micro-pitch racks

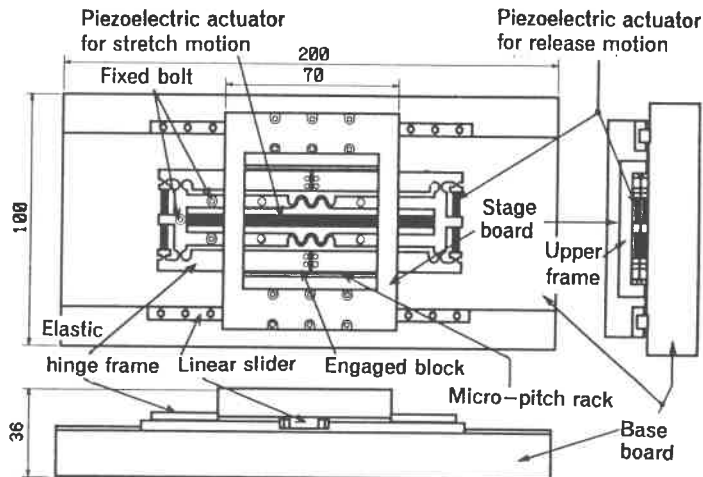


Fig. 2 Structure of the precise positioning stage

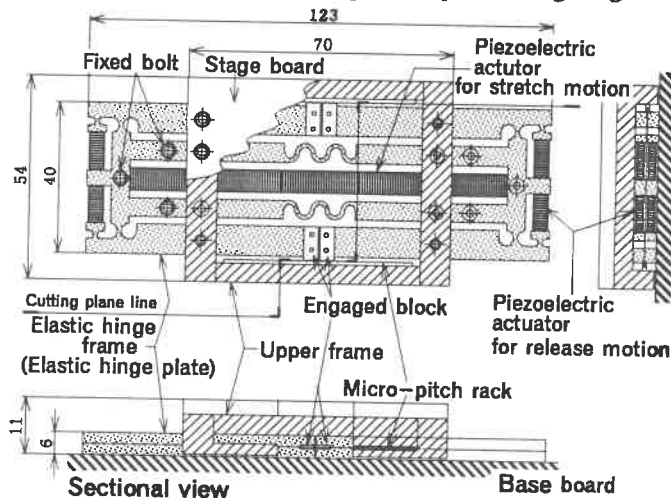


Fig. 3 Structure of the elastic hinge frame

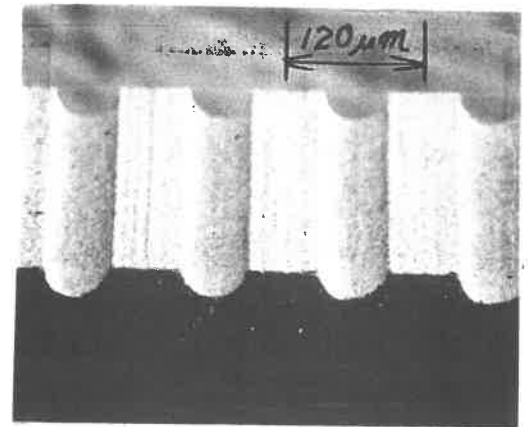


Fig. 4 Teeth profile of the micro-pitch rack

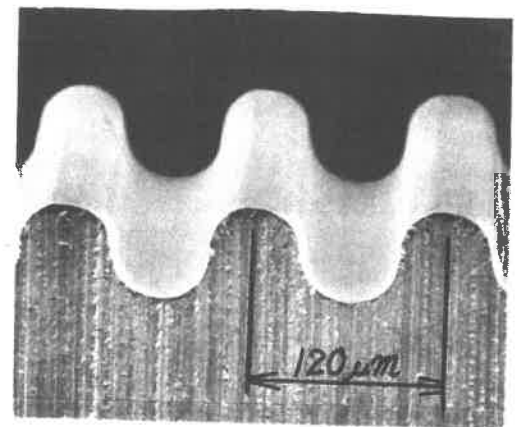


Fig. 5 Teeth profile of the engaged block

Two micro-pitch racks facing each other are attached under the stage board. Fig. 3 shows details of the elastic hinge frame and location between the micro-pitch racks and the elastic hinge frame. The elastic hinge frame consists of two elastic hinge plates. Each elastic plate has four arms and two pre-load springs. Four engaged blocks which have the same micro-pitch teeth are attached at those tips of the arms respectively, and four piezoelectric actuators are installed in edge spaces of the arms. When a pair of upper and lower piezoelectric actuators is extended, a pair of upper and lower engaged blocks is released from the micro-pitch racks. As the elastic hinge frame is composed of two elastic hinge plates, four pairs of the engaged blocks are released by turns from the micro-pitch racks. Between the pre-load springs, another piezoelectric actuator is installed to extend an interval of the engaged blocks. The extension is used in the inchworm motion.

Thus four pairs of the engaged blocks mesh alternately with the micro-pitch racks in the procedure of the inchworm motion. The stage board is led every 60 μm over 40 mm. The stage is 200 mm in length, 100 mm in width depth and 36 mm in height. The stage board is 70 by 80 mm.

The teeth profile of the micro-pitch rack and the engaged blocks are shown in Fig. 4 and Fig. 5. The micro-pitch rack is made of special steel for dies (JIS: SKD11), and depth of grooves is 30 μm in radius. The engaged block is made of cemented carbide (JIS: WC3), and section profile of the teeth is a series of arcs which combined semicircles of 30 μm in radius. Both pitches of the micro-pitch rack and the engaged block are 120 μm.

3. Travel sequence of the stage board for the inchworm motion

Relative location of the micro-pitch rack and four engaged blocks is shown in Fig. 6. The micro-pitch teeth of four engaged blocks are shifted half pitch of the micro-pitch rack. In this stage, the stage board is driven by using following drive sequence of the piezoelectric actuators. The drive sequence is schematically shown in Fig. 7.

(1) Regular position of the stage board

When the piezoelectric actuator for stretch motion does not extend, the engaged blocks of A1, A2, D1 and D2 mesh with the micro-pitch racks and the engaged blocks of B1, B2, C1 and C2 are released from the micro-pitch rack.

(2) Stretch motion of the elastic hinge frame

The engaged blocks of D1 and D2 are released, and the piezoelectric actuator for stretch motion is extend half pitch of the micro-pitch rack. The engaged blocks of C1 and C2 come to the position to mesh with the micro-pitch racks as the engaged blocks of C1, C2, D1 and D2 are moved at the same time.

(3) Exchange motion of the engaged blocks

After the engaged blocks of C1 and C2 mesh with the micro-pitch racks, the engaged blocks of A1 and A2 are released from the micro-pitch racks.

(4) Traction motion of the stage board

When the piezoelectric actuator for stretch motion contracts to the normal length, the stage board is led half pitch of the micro-pitch rack to the left of the elastic hinge frame because left side of the elastic hinge frame is fixed on the base board.

(5) Sub-regular position of the stage board

The engaged blocks of B1 and B2 mesh with the micro-pitch racks as the engaged blocks of B1 and B2 come to the position to mesh with the micro-pitch racks. The condition in which the engaged blocks of B1 and B2 mesh with the micro-pitch racks is an inverse condition of the regular position. The half sequence for the inchworm motion is concluded.

After several repeats of the half sequence mentioned above, the stage board is positioned to a reference position in the procedure of the inchworm motion composed of the open-loop control system.

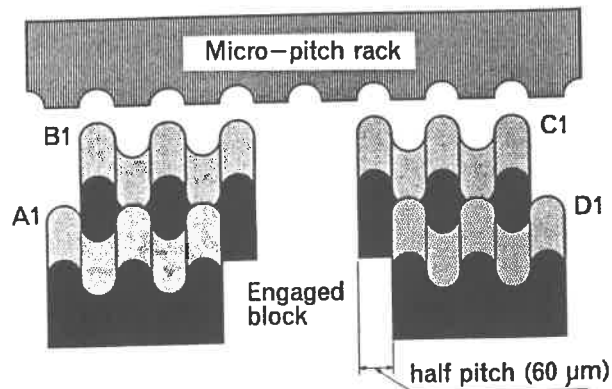


Fig. 6 Relative location of the micro-pitch racks and four engaged blocks

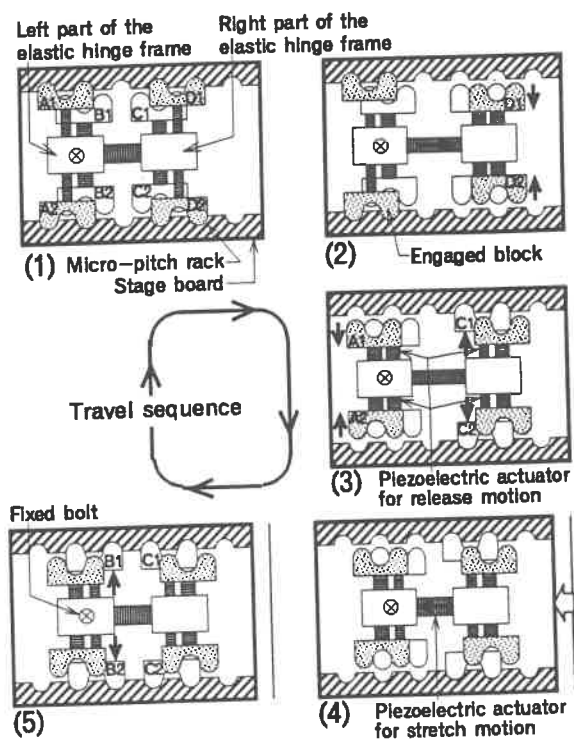


Fig. 7 Travel sequence for the inchworm motion

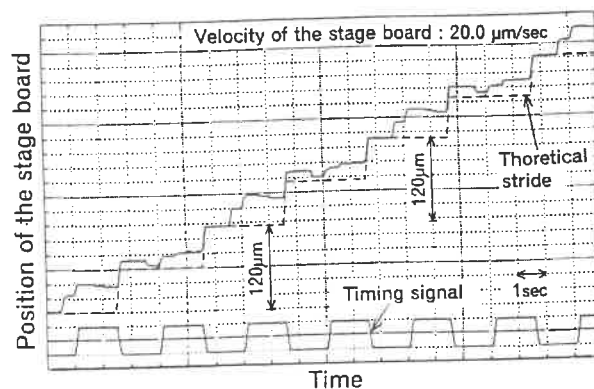


Fig. 8 Stepwise positioning of the stage board

4. Experimental results

In the experiment, position of the stage board was measured by a non-contact gap sensor.

Stepwise positioning of the stage board was shown in Fig. 8. In this figure, when a timing signal changed from high level to low level, that is, when the piezoelectric actuator for stretch motion contracted to the normal length, the stage board was led about $60\text{ }\mu\text{m}$ step by step. After quarter cycle of the drive sequence, the stage board ran over $20\text{ }\mu\text{m}$ or moved back $10\text{ }\mu\text{m}$. The deviation of the positioning was caused by assembly errors of the engaged blocks because the deviation occurred when the engaged blocks meshed with the micro-pitch racks. However, distance between the start position and the end position of every whole cycle was correspond to the theoretical stride of $120\text{ }\mu\text{m}$ as shown in Fig. 8. This accurate stride was caused by cancellation of pitch error because the engaged block was in contact with the micro-pitch rack at a lot of teeth and error of the stride was averaged nearly to zero. Therefore, this stage can position every one pitch distance of the micro-pitch rack even if the micro-pitch rack is manufactured with pitch error.

Trace curves on which the stage board were moved five strides with different velocity were shown in Fig. 9. In this figure, if the repeated number of the drive sequence was equal, the stage board was positioned at the same position. This meant the stage was positioned without slipping in the inchworm motion. The repeatability of the stage board was shown in Fig. 10. In this figure, the stage board moved forward ten strides and moved back ten strides and the round positioning was repeated eight times. The start position and the end position did not deviate from those positions. Positioning errors of the stage board were shown in Fig. 11. In this figure, the maximum error of the theoretical stride was $1.30\text{ }\mu\text{m}$, the maximum error of the accumulated stride was $2.3\text{ }\mu\text{m}$ and the maximum error of the adjoining stride was $2.5\text{ }\mu\text{m}$.

5. Conclusions

Novel precise positioning stage traveling on the micro-pitch racks is developed. The stage is positioned by meshing the engaged blocks to the micro-pitch racks in the inchworm motion. The stage can accurately position every whole pitch of the micro-pitch rack without slipping.

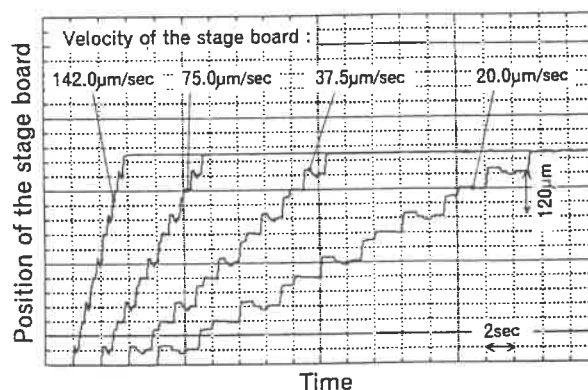


Fig. 9 Trace curves with different velocity

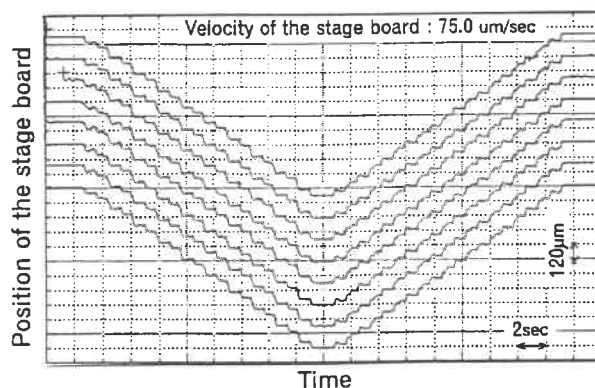


Fig. 10 Repeatability of the stage board

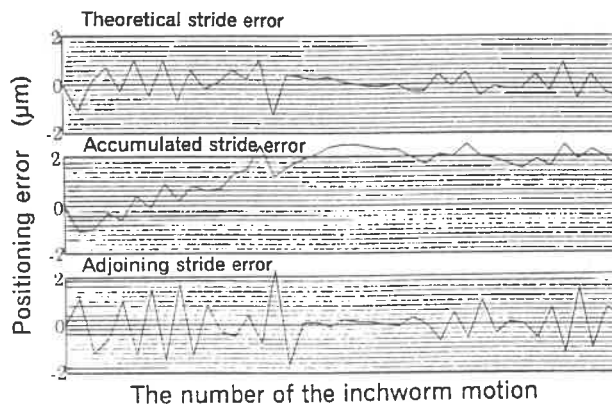


Fig. 11 Positioning error of the stage board

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Surface Self-Compensated Linear and Rotary Hydrostatic Bearings

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Novel linear and rotary hydrostatic bearings are presented which are well performing while being ideally suited for mass production due to the minimal number of parts they require. These bearings achieve stiffness by a surprisingly simple means: each load bearing pad is fed by an auxiliary throttling surface which make an acute angle relative to the bearing pad it feeds. The angled throttle surface design gives good static stiffness feedback and eliminates the need for extra parts such as capillaries or small orifices.

The current linear prototype is bolt for bolt compatible with size 35 off the shelf linear guideways currently on the market, and it utilizes the same profile grinding technology used for rolling element guideways. The new design consists of a profiled rail and a conforming one piece runner block. For its size, the measured stiffness and load capacity is acceptable for many machining and grinding applications (one size 35 truck running at 70 Bar pressure gives a load capacity of 460 kg, a compressive stiffness of 420 Newtons/micron, and a lateral stiffness of 250 Newtons/micron). The major advantages the new hydrostatic design can provide over existing rolling elements are infinite life at high speeds, better damping, and smoother, more repeatable motion. The main applications are precision grinding machines and high speed machining centers which use linear motors.

The current rotary prototype (250 mm O.D., 90 mm tall) consists of only five precision round pieces, and provides a radial error motion of 0.13 micro meters using parts which are round to 2.5 micro meters. This 20 to 1 error reduction is a great improvement over rolling element bearings, which typically achieve the same error motion as the roundness of the parts. Compared to existing aerostatic and hydrostatic designs having both the same size and operating pressure, the new design provides similar error averaging while having both a lower cost and a higher load capacity. The new design is scalable to any size and is an ideal replacement for large rolling element or sliding rotary tables (600mm diameter or larger) for machine tools. For its size, the measured stiffness is acceptable for most machining applications (running at 40 Bar pressure the radial stiffness is 1000 Newtons/micron and the axial stiffness is 1600 Newtons/micron).

Shakedown Characterization of an Instrument for Nanocutting and Nanoindenting

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Introduction

Even though there has been an increase in the study of the ultra-precision machining process, there still exists a need to further our understanding of the basic process physics when cutting in the sub-micrometer and nanometer uncut chip thickness range. Commercial diamond turning machines have been used to investigate sub-micrometer depths of cut but, experimental work [1] indicates that both control of depth of cut and measurement of process forces become questionable on a commercially available diamond turning machine when cutting in the nanometer range. A bench instrument, which is capable of controlled nanoindenting and nanocutting, was designed and built. This paper presents initial tests that were performed on the instrument to ascertain its capabilities.

Instrument Design

The instrument, described further elsewhere [2,3], is composed of an aluminum box frame measuring 200 x 125 x 75 mm and contains two modules inside the frame, viz., the force tripod and horizontal slide. A piezoelectric tube drives the sample into a stationary indenter or across a stationary tool. There is optical access to the cutting or indenting area from the top and side of the instrument. Normal and tangential forces on the tool can be measured. The instrument is designed to achieve uncut chip thicknesses from 2.5 μm down to several nms, and to be vacuum compatible. The cutting speed can be varied up to 0.3 m/min.

The force tripod houses the diamond indenter or cutting tool and the force transducers, see Fig. 1. The tool is mounted in the tool holder with the rake face normal to the horizontal plane of the instrument with cuts being made in the X-direction. The tool holder is held in place by dual-axis flexures which are designed to decouple the thrust and cutting forces. The stiffness of the dual-axis flexures is 30 MN/m in the axial direction and 0.3 MN/m in the off-axis direction. The off-axis stiffness was chosen so that the natural frequency of the force tripod is 2.5 kHz, which is higher than the expected maximum frequency (0.5 kHz) in operation. The cross coupling between the cutting and thrust force transducers has been measured to be less than 5%. Two single component force transducers with a range of 0.5 mN to 20 N and a resolution of 0.2 mN are used. The transducers have a preload of about 10 N. The entire force tripod can be coarse

positioned in the vertical direction by a hysteritic drive. The drive can be decoupled after positioning the force tripod to take it out of the structural loop. The horizontal slide is a single-axis flexure housing the piezoelectric tube. The piezoelectric tube is 25.4 mm long and 12.7 mm in diameter, and has an axial stiffness approximately equal to that of the dual-axis flexures, and flexural stiffness approximately one order of magnitude less. The sample is glued to a sample mount holder, which is kinematically mounted to the piezoelectric tube. A Maxwell (three V-groove) coupling designed to have a stiffness of 7.5 MN/m is used to hold the Macor sample mount to the piezoelectric tube. The horizontal slide can be coarse-positioned in the Z-direction over a range of 7 mm by an 80 TPI screw. For fine-positioning in the Z-direction, the piezoelectric tube can move the sample over a range of 3 μm with a resolution of 0.1 nm. The piezoelectric tube also provides 11 μm cutting motion in either of the planar directions with a resolution of 3 nm. A capacitance gage is mounted inside of the piezoelectric tube to measure the displacement of the sample in the Z-direction. The effective rms noise due to A/D conversion and capacitance gage electronics in reading displacement was found to be 0.3 nm.

To establish the sample surface, the sample is oscillated sinusoidally at 235 Hz and advanced toward the tool or indenter with a user-chosen step size. This frequency was determined by exciting the sample with a random signal and monitoring the thrust force output. The thrust force output was flat to within one percent up to 240 Hz. A 235 Hz signal is used as the reference signal for a lock-in amplifier. To prevent any permanent deformation of the sample, the sample is moved toward the tool in 2 nm steps. The sample is retracted from the tool by 75 nm once the lock-in amplifier detects an output voltage from the force transducer.

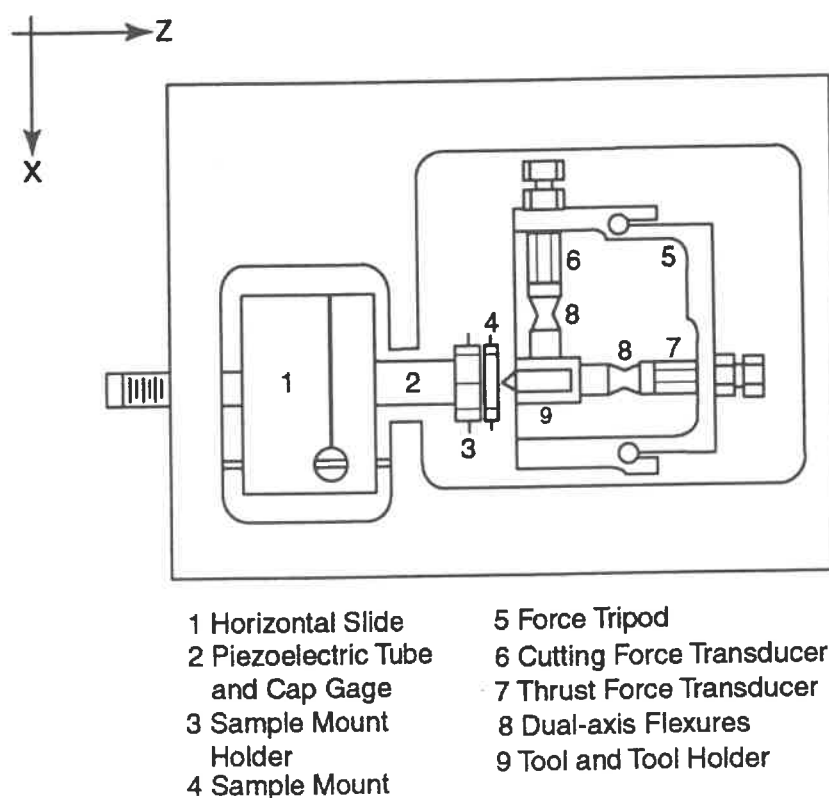


Figure 1: Schematic view of the instrument.

Evaluation of Instrument

As mentioned above, the frequencies over which the instrument can be operated were determined by exciting the sample with a random signal and measuring the output of the thrust force transducer. The output of the thrust force transducer was flat to within 5% up to 900 Hz, which is above the estimated maximum operating frequency of 500 Hz. The force transducers were calibrated *ex situ* by deadweights over a range of 5.2 mN to 490 mN. The voltage versus force curves were linear with an R^2 value greater than 0.995. The capacitance gage was calibrated using a laser interferometer. A Berkovich indenter was used in the experiments and the geometry of the indenter tip was measured with an atomic force microscope (AFM). The cantilever tip radius of the AFM was determined [4] and the AFM image of the indenter was deconvolved [5] using this tip radius. The area function of the indenter, which relates the depth measured from the tip of the indenter to the projected area of the indenter, was determined from the deconvolved AFM image of the indenter.

The piezoelectric transducers have a time constant such that the signal loss is less than 5% over the first 100 ms. To minimize the errors associated with the signal decay, the indentations consist of a series of steps and the output of the transducer is recorded with an oscilloscope. A simple feedback loop is used to maintain the depth of the steps. To ensure that the unloading of the sample is elastic, multiple loadings are performed, as shown in Fig. 2. After the location of the sample surface has been established, the sample is indented to 85% of the desired indentation depth and then withdrawn to 50% of the desired depth. This process is repeated a second time. On the third and final loading the sample is indented to the desired depth and the sample is then totally disengaged from the indenter.

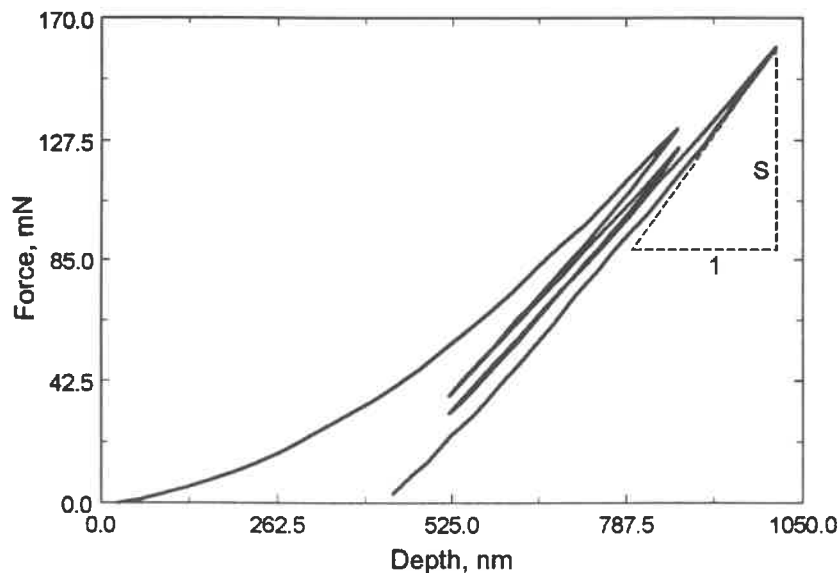


Figure 2: Indentation in fused silica with a maximum depth of 1010 nm. The slope of the final unloading curve, S , is the inverse of the total measured compliance.

Indentations in fused silica were made with a nominal depth under load of 800 nm, 1000 nm, 1200 nm, and 1400 nm. One set of experiments consisted of four indentations at each of the four depths and three sets of experiments were made. The entire final unloading curve was fit to a power relationship, which was analytically differentiated and evaluated at the maximum depth to determine the total stiffness [6]. The inverse of the total stiffness is the total compliance. The

instrument and the sample can be modeled as two springs in series so that the total measured compliance is just the summation of the compliance for each element. In Fig. 3, the total compliance is graphed with respect to the contact area^{-1/2}. Since the compliance of the instrument is independent of the contact area, the intercept of the compliance axis is the compliance of the instrument (frame compliance), about 0.3 nm/mN. This is similar to the compliance of commercially available nanoindenters [7].

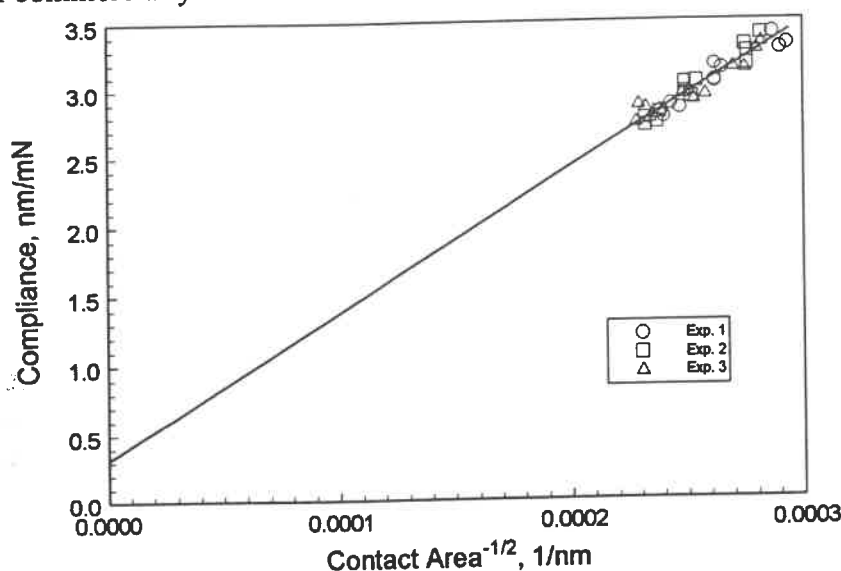


Figure 3: Total compliance vs. contact area^{-1/2}. The frame compliance is the intercept of the compliance axis, about 0.3 nm/mN.

Acknowledgments

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DESIGN OF A THERMAL GRADIENT ATTENUATOR

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Introduction

A recirculating air chamber has been constructed with the purpose of attaining milliKelvin levels of temperature control. Chilled water is pumped through a cross-flow heat exchanger at varying rates to maintain constant air temperature exiting the heat exchanger. The processed chilled water, pumped through a cross-flow heat exchanger, has a wide spectrum of temperature oscillations (Figure 1 and 2) that cause disturbances in the air temperature. Because of the pump and motor inertia, high frequency water temperature oscillations cannot be sufficiently attenuated to avoid unacceptable air temperature deviations. To address this problem, we have designed a thermal gradient attenuator that reduces water temperature gradients to suitable levels.

The optimum thermal gradient attenuator was constructed with a 55-gallon drum and 2500 feet of 3/16" diameter nylon tubing. 947 nylon tubes are held in a hexagonal pattern with 4 acrylic plates. The water to be attenuated flows through the tubes which are submerged in the water-filled drum. The stagnant water in the drum attenuates the thermal gradients in the water flow.

This design was chosen on the basis of optimal gradient attenuation and cost. Water is an inexpensive and efficient attenuator material because of its availability and high heat capacity. The optimal blend of thermal resistance from the flowing water to the attenuator water and the associated thermal capacitance of the attenuator water was computed based on measured data of the water supply to be attenuated.

The following integral was minimized in the design of the absorber to minimize the gradients exiting the gradient attenuator.

$$M = \int_{f_{\min}}^{f_{\max}} |\text{Tr}(f)| \cdot |\text{grad}(f)| \cdot df$$

where: $\text{Tr}(f)$ - transfer function of the attenuator
 $\text{grad}(f)$ - input water gradient as a function of frequency
 $f_{\min}$ - minimum frequency for significant gradients
 $f_{\max}$ - maximum frequency for significant gradients

Theoretical Model

The theoretical model for the attenuator performance is based on the heat flow from the flowing water to the stationary attenuator elements, as shown in Figure 3. The attenuator heat equation can be expressed as:

$$\frac{1}{R} \cdot (T_x - T_a) = C_a \cdot dx \cdot \frac{\partial T_a}{\partial t}$$

where: R - thermal resistance between water and absorber elements
 T_a - absorber temperature at midpoint of differential element
 T_x - water temperature at midpoint of differential element
 C_a - heat capacity per unit length of absorber material
 dx - differential length

The water flow heat equation (neglecting axial conduction) can be expressed as:

$$\dot{m}_w \cdot c_w \cdot (T_i - T_o) + \frac{1}{R} \cdot (T_a - T_x) = C_w \cdot dx \cdot \frac{\partial T_x}{\partial t}$$

where: $\dot{m}_w$ - mass flow rate of the chilled water
 C_w - heat capacity per unit length of water
 T_i - water inlet temperature
 T_o - water outlet temperature
 R - thermal resistance from water to absorber (K/W)
 T_a - absorber temperature at midpoint of differential element
 T_x - water temperature at midpoint of differential element

After transforming to frequency space with the derivative operator, s , and substituting the absorber equation into the water equation, the transfer function can be expressed in the following form:

$$Tr(s) = \frac{T_w(u = u_f, s)}{T_w(u = 0, s)} = \exp \left[\frac{-s}{s+1} \cdot (s + \zeta + 1) \cdot u_f \right]$$

where: $u_f = \frac{L}{v \cdot \tau_a}$
 $\zeta = \frac{c_a \cdot \rho_a \cdot A_a}{c_w \cdot \rho_w \cdot A_w}$ - relative 'stiffness' of the absorber and the water
 $u = \frac{x}{v \cdot \tau_a}$ - dimensionless length unit
 $\tau = \frac{t}{\tau_a}$ - dimensionless time unit
 v - flow velocity of the water

Experimental Results

The attenuator was tested using the setup depicted in Figure 4. A vane pump maintains a constant flowrate of water through the system. The computer controls power input to a cartridge heater via a data acquisition board and variable power supply to force a sinusoidal temperature deviation in the water flow. The computer reads the thermistor resistances from the high-precision digital multimeter via GPIB interface. The chiller and heat exchanger remove the nominal heat load from the system.

The measured performance of the attenuator matches the theoretical performance at a flowrate of 0.67 gallons per minute, with less than 3 dB RMS deviation (Figure 5). In the region of the worst gradients, the attenuator performance is approximately one order of magnitude better than the theoretical attenuation of perfectly mixed water in a 55-gallon drum (Figure 5). In the frequency range of 0.5 to 2 mHz the tube attenuation is several orders of magnitude better than that of thermal diffusion. Above 3 mHz the tube attenuator loses its performance advantage, but the difference in attenuation is unimportant since the attenuation is already 3 orders of magnitude.

At present, the theoretical models predict higher sensitivity to changes in flowrate than the measured data demonstrates. The theoretical transfer functions predict the same overall behavior of the performance, but the method of computing the overall thermal resistance between the water flow and the attenuator elements does not accurately predict the asymptotes of the measured data at all flowrates. We are in the process of taking more transfer function data at different flowrates and investigating numerical models to determine the source of the asymptote discrepancy.

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Figure1: Chilled Water Temperature and Instantaneous Derivative vs. Time

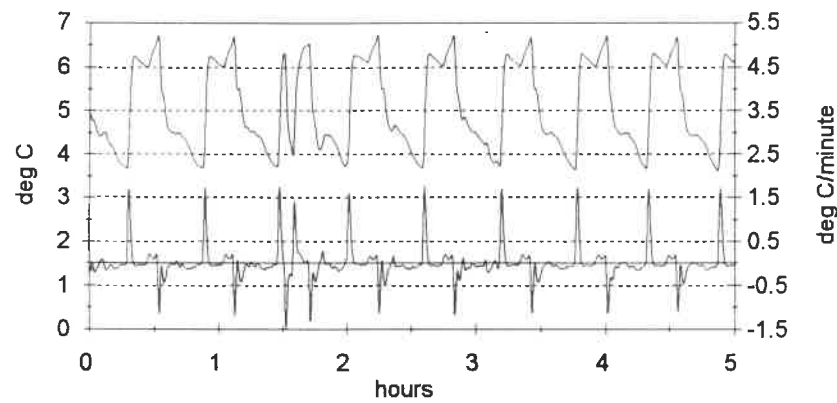


Figure2: Chilled Water FFT Average Temperature Derivative vs. Frequency

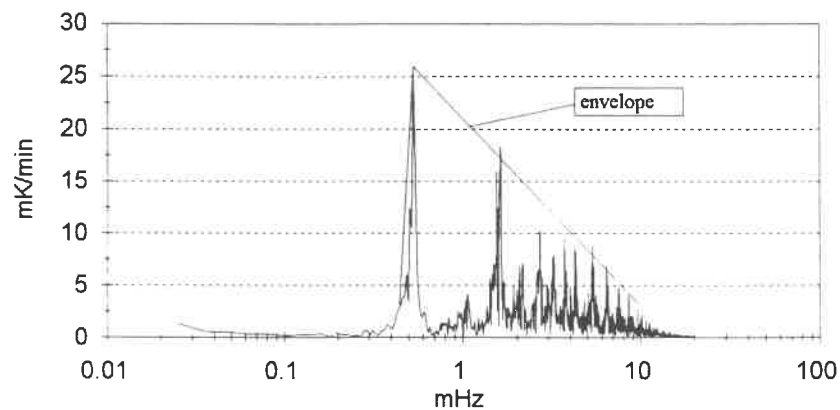


Figure 3: Differential Elements in Heat Flow Equations

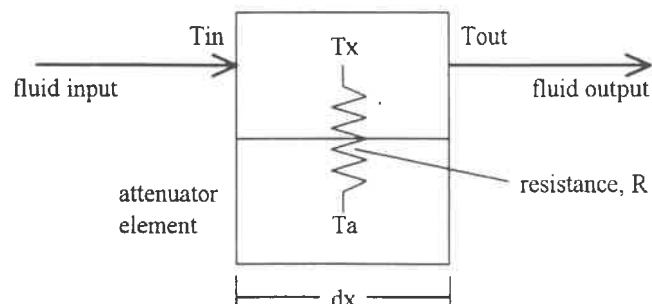


Figure 4: Attenuator Test Setup

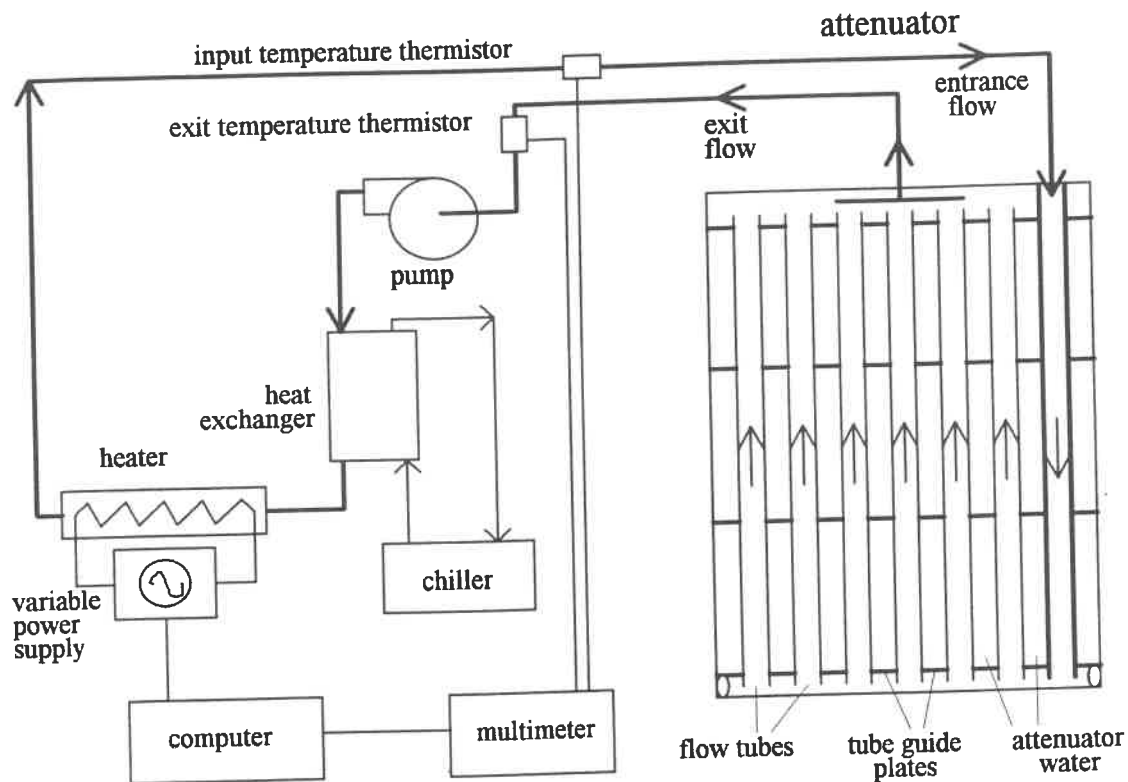
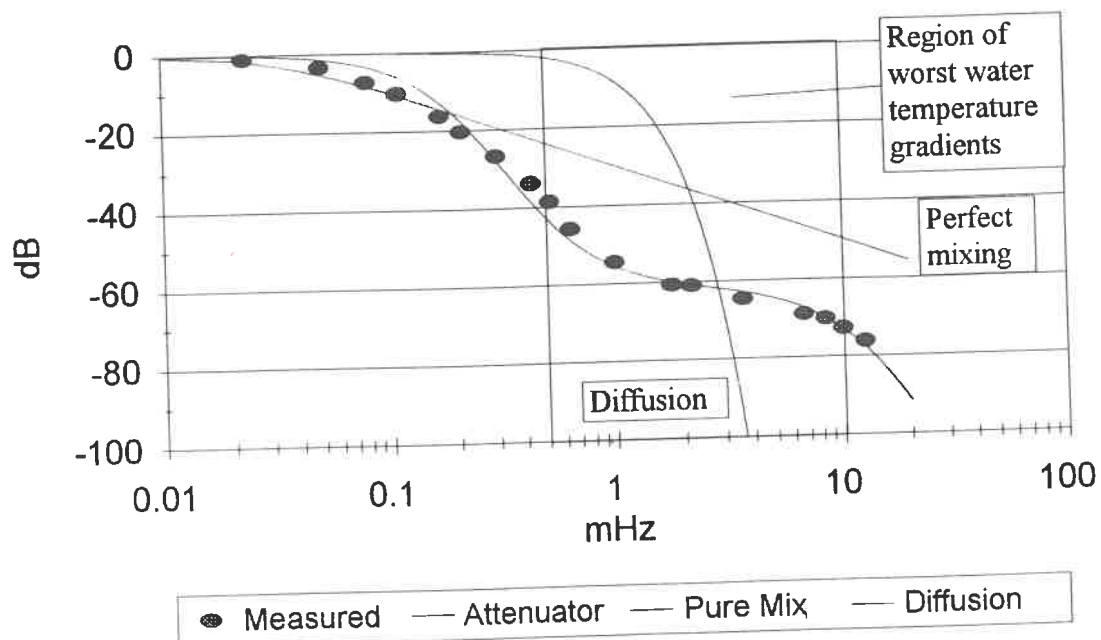


Figure 5: Drum Transfer Function
2/3 gpm: Attenuator, Mixing, Diffusion



Pushing the Thermal Limit in Linear Motors

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Abstract

We present the design of a high force per unit volume linear motor for use in machine tools. The motor incorporates coils wound with separated end-turns so that each layer of the coils can be directly cooled. Oil flows through the gaps in the end-turns on both sides of each coil to remove heat. A current of 1.6 A causes a 100°C temperature rise in a free convection-cooled coil; it takes a current of 9.0 A to cause the same temperature rise with our cooling technique. Thus our design allows nearly 6 times higher force in steady state and dissipates 32 times as much heat.

Introduction

A cross-section of a typical permanent magnet synchronous linear motor is shown in Figure 1. The magnets are mounted on a U-shaped piece of back iron and create an alternating magnetic field in the air gap. A moving electrical coil assembly containing three rectangular coils labeled A, B, and C rides in the air gap. The end-turns of each coil (not shown) are outside of the magnetic field and hence don't contribute to the motor's force. The force produced is proportional to both the magnetic field and the current in the coils. The magnetic field strength is determined by geometry and type of permanent magnets used. The maximum current allowed is limited by I^2R Joule heating in the coils. This thermal limit is one of the biggest obstacles to increased linear motor performance [1, 3].

A thermally efficient motor produces more force in a given volume before hitting the thermal limit. It allows machine tool designers to use fewer or smaller motors while still producing the required forces. It can also allow for cooler operation which is advantageous for three reasons. First, since coil resistance increases 40% for a 100°C temperature rise, it is more power efficient to operate a motor at lower temperatures. Second, the motor will have a longer life if run at lower temperatures because insulation materials degrade twice as fast for every 10°C rise in temperature. Third, thermal expansion errors are reduced. Thus, improved motor thermal efficiency allows for

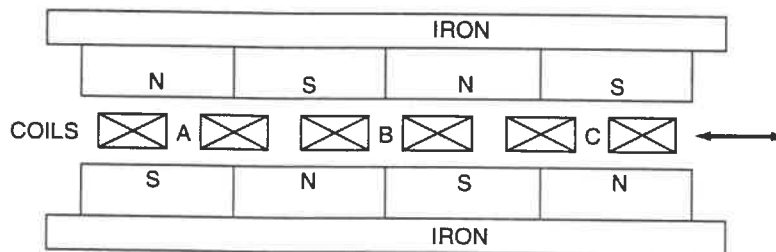


Figure 1: A cross-section of a permanent magnet synchronous linear motor is shown. The non-overlapping coil design [2] improves the packaging.

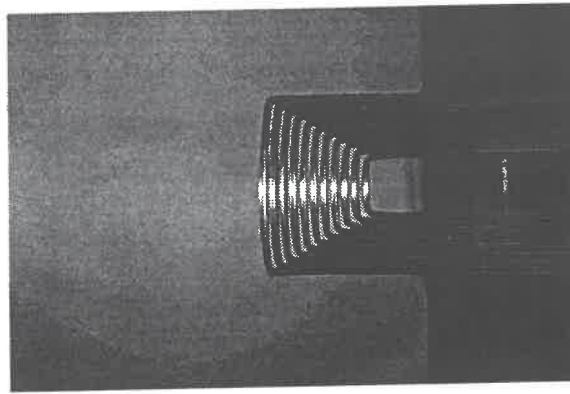


Figure 2: One half of the separated end-turn coil design is shown. The coil consists of 22 layers which are separated into 11 groups of two at both end-turns. Grouping them in twos cut down the end-turn size while still allowing each layer to be directly cooled.

increased power efficiency, increased maximum motor force, decreased size, longer life, increased machine accuracy or a combination of these.

Conventional cooling techniques cool only the outermost layer of wires in a coil by, for example, placing a copper tube with flowing air or water next to the coil's end-turns. The heat generated in the many interior layers must travel across the outer coil layers to escape. The thermal resistance across coil layers is substantial, leading to unacceptably high temperature gradients in the coil. A simple thermal analysis [4, 5] shows that the temperature rise across a coil goes as the number of coil layers squared. Conventional cooling techniques afford moderate increases in current and hence force: Conventional forced air cooling of linear motors typically allows 1.15 times the force of a free convection-cooled motor while conventional forced water cooling typically allows 1.35 times the force. A better cooling scheme must cool each layer of the coil directly. The scheme presented next does this and has demonstrated 5.6 times the force of a free convection-cooled motor in steady state.

Coil Design

To avoid the problem of cooling across multiple layers, we have designed a coil with gaps in the end-turns. The main working part of the coil (shown in cross-section in Figure 1) is the same, but the gaps in the end-turns allow each layer of wires to be directly cooled. This separated end-turn coil is shown in Figure 2. Heat is generated everywhere in the coil. It is conducted down the length of each wire to the end-turn where it is removed directly by a heat sink placed in the gaps. This heat sink can be either a flowing fluid such as air or oil, or it may be a solid such as copper. In this fashion we are no longer limited by the large inter-layer thermal resistances in the coil.

The thermal model we have developed for the separated end-turn coil divides the coil into two regions [4]. The first region consists of the main working part of the coil which is uncooled, and the second region consists of the cooled end-turn. Depending on the application, the temperature rises across these two regions might be designed to be approximately equal, or one might dominate the other. The geometry of the coil, type of coolant, and flow rate of coolant control the relative temperature rises across these two regions. For example, the temperature rise from the end-turn

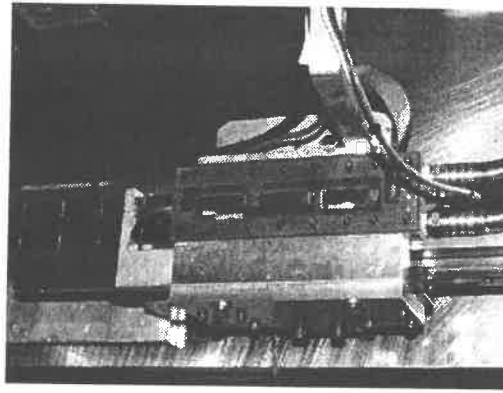


Figure 3: A top view of our prototype linear motor is shown. The upper housing of the three coils contains baffles which direct the oil through the coils' separated end-turns shown in Figure 2.

to the center of the coil goes as the length of wire squared and the current squared. A current of 8 A causes a temperature rise of 100°C from the end-turn to the center of a 12 cm long coil. Thus even if the temperature rise from the coolant to the end-turn is negligible, the temperature rise along the coil is prohibitive, and we could not expect to exceed this current level with a coil of this length.

Experimental Results

The first cooling technique we describe is direct liquid cooling of the separated end-turn coils. Oil flows directly through the gaps in the coils' end-turns. We have designed and built a prototype permanent magnet, synchronous linear motor which incorporates this direct oil cooling as shown in Figure 3. Our prototype motor has the geometry shown in Figure 1. It has a force constant of $35.3 \text{ N/A}_{\text{rms}}$ and a peak magnetic field of 0.75 T. With liquid cooling, it produces a force of 259 N while dissipating 730 W with a peak current density of $3.47 \times 10^7 \text{ A/m}^2$ in the wire.

A current of 1.6 A causes a steady-state 100°C temperature rise in a free convection-cooled coil like the one shown in Figure 2; it takes a current of 9.0 A to cause the same temperature rise in our motor with room temperature oil flowing through the end-turns at one-third gallon per minute. Thus, our design allows 5.6 times higher force in steady-state while dissipating 32 times as much heat. The associated thermal test data is plotted in Figure 4. Notice that at a current of 2.0 A, the average temperature rise in the oil-cooled motor is only 8°C , whereas the free air convection-cooled motor overheats at this current. Furthermore, a free convection-cooled coil will stay hot for 30 minutes after the current is turned off; with oil cooling, the hot spot temperature falls from 130°C to 44°C in 30 seconds. The coil's thermal time constant is approximately 60 times faster than a free convection-cooled coil and approximately 20 times faster than a conventional coil with air blowing on it.

The second cooling technique is called "comb-cooling" because a comb-shaped piece of copper is placed into the separated end-turns of the coil. Heat generated in the coil leaves the end-turns and flows through the comb's copper fingers to its base. From there, it is removed by flowing water. Since the coolant does not directly contact the coils, the comb-cooling circulation system is much simpler to manufacture than in direct liquid cooling. Our first comb-cooling prototype is crude

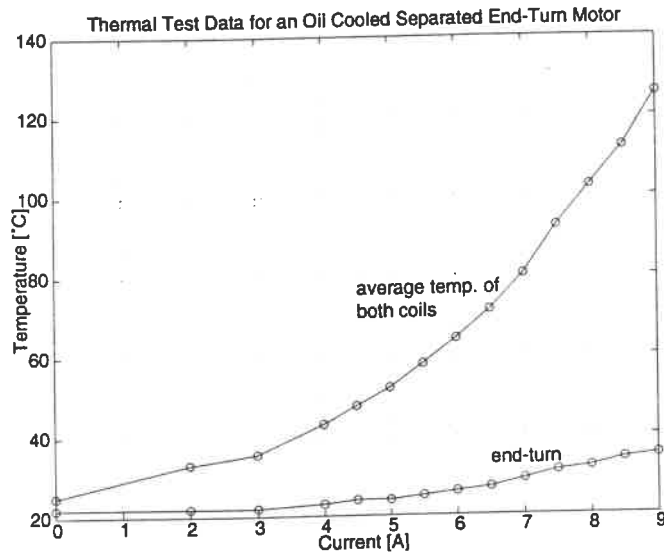


Figure 4: A DC current was applied across two of the three coils in our motor while oil circulated through the end-turns at a flow rate of 0.3 gpm. The average temperature of the two coils is plotted versus current. Also shown is the temperature at the middle of the end-turn. When uncooled, this same coil heats up to a nearly uniform 125°C with only 1.6 A.

since it is only implemented on one end-turn, the fingers are thinner than the gaps in the coil, and it contains low thermal conductivity epoxy. Nonetheless, this comb allows 2.4 times the current of a free convection-cooled coil. With improvements in fabrication, we predict that this technique would allow 4.7 times the current of a free convection-cooled coil and dissipate 22 times the power.

Acknowledgments

Our partnership with Anorad Corporation not only provided funding for this research, but allowed us access to their knowledgeable staff and manufacturing techniques. We are particularly grateful to Anwar Chitayat for his innovative ideas and Fred Sommerhalter for machining the prototype motor parts. Portions of this work are from a thesis submitted by Michael Liebman [4] for the Master of Science in Mechanical Engineering at Massachusetts Institute of Technology.

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The Dimensional Stability of Lightly-Loaded Epoxy Joints

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Abstract

The use of adhesives to bond metal mounting structures to optical components can significantly simplify the design of an optical system. In precision applications, dimensional instability of the adhesive must be included as a component of the overall error budget. This paper describes the qualification testing of a balanced heterodyne interferometer system in a carefully controlled environment for the purpose of measuring joint stability. Results of this qualification test are reported.

Introduction

Advanced imaging systems, such as those required by extreme ultraviolet lithography require optics figured with subnanometer accuracy over large apertures and must be accurately located within a few nanometers. The preservation of the surface figure and the dimensional stability of the optic's surface relative to the other optics dominate the performance requirements of the optic's mounts.

One concern when using an epoxy in such applications is its long-term stability along the optical axis (shear microcreep). The long-term error budgets for epoxy creep must be kept to a small portion of the total error budget, which may be on the order of 1 micron for each optic. To measure the creep rate of the epoxy, a high-resolution displacement measuring interferometer mounted in a temperature controlled chamber is used to measure the creep of a lightly loaded epoxy joint. In order to verify that the inherent system stability is adequate, a balanced arm heterodyne interferometer has been constructed and used to determine a lower bound on the uncertainty of the creep measurement. The experimental setup, environmental control, and preliminary results are described in this paper. Ultimately, it is desired to

demonstrate the sensitivity required for monitoring test samples for short periods of time, spanning many days, or possibly weeks, to establish an estimate of long term creep rates.

Experimental Setup

The setup used for determining the base stability of the system is a four-pass plane mirror interferometer. Figure 1 shows a schematic of the optical arrangement.

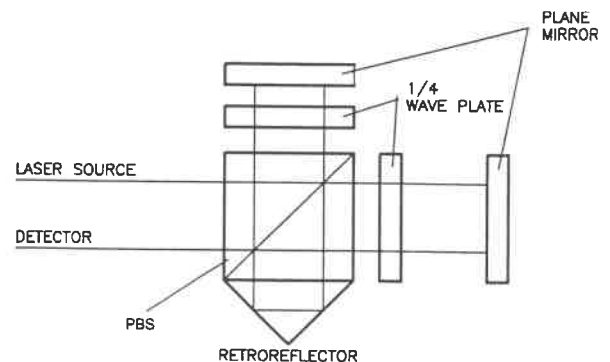


Figure 1: Optical Schematic of the Interferometer

It consists of nominally identical tubular aluminum spacers in each arm that support the plane mirror at a nominally fixed distance from the polarizing beamsplitter (PBS). The spacer is mounted to the body of the PBS using a semi-kinematic three-point contact. The mirror is a $\lambda/10$ aluminized low expansion two-phase glass ceramic optical flat which is mounted using a semi-kinematic three-point mount that contacts the face of the mirror. The spacers are attached to the PBS body with spring-loaded threaded fasteners in order to provide a constant-force joint. Belleville washers provide the spring preload. The retroreflector is similarly attached. The

mirror is preloaded against its three-point mount by a leaf spring. To minimize the influence of air property changes during the measurement, care has been taken during assembly to ensure that the optical path difference between the two legs of the interferometer is less than 20 microns. The squareness of the faces of the PBS to which the spacers attach is carefully controlled along with the parallelism of the mirror faces to the mounting flanges of the spacers in order to minimize errors due to beam shear. Figure 2 shows the various components in the assembly.



Figure 2: Interferometer Assembly Components

Measurement Method

The interferometer is mounted to a large aluminum block and placed in a precision temperature control chamber. The chamber temperature control consists of a chiller heat exchanger followed by an electric reheat coil.

The chiller evaporator coils are controlled by a thermostatic valve mounted directly to the evaporator coils. Precision temperature control is achieved by using three cascaded control loops to drive the reheat coils. Feedforward from the chiller control to the reheat coils is also included to increase rejection of variations in the chiller performance. Each temperature feedback loop uses a precision glass bead thermistor. We have shown previously that these thermistors are stable at the millidegree level over times greater than those involved with the creep measurement [1].

The temperature control chamber typically maintains short-term fluctuations below 6 mK and long term variations within a ± 1 mK band at the table in the vicinity of the interferometer. The interferometer phase interpolation electronics are also

mounted in the chamber. Temperature control at the electronics is typically ± 12 mK, with most of the variation due to a diurnal cycling of the power line voltage and the consequent variation in heat dissipation.

During measurements, in addition to the air temperature, both the temperature of the interferometer body and the atmospheric pressure in the chamber are monitored. Some measurement bias may be removed by fitting the dimensional data to temperature and pressure values. Since the observed temperature and pressure variations over the length of several days of measurement contain fluctuations that are significantly larger than their long term trends, fitting and subsequently removing small residual pressure and temperature effects does not appreciably increase the uncertainty of the creep measurement.

Measurement Results

Figure 3 depicts the observed dimensional change of the reference interferometer assembly. After an initial settling period following the closure of the temperature control chamber, the reference interferometer position is asymptotic to a line with a slope of 0.4 nm/day. This value is taken as a lower bound on the uncertainty of creep measurement.

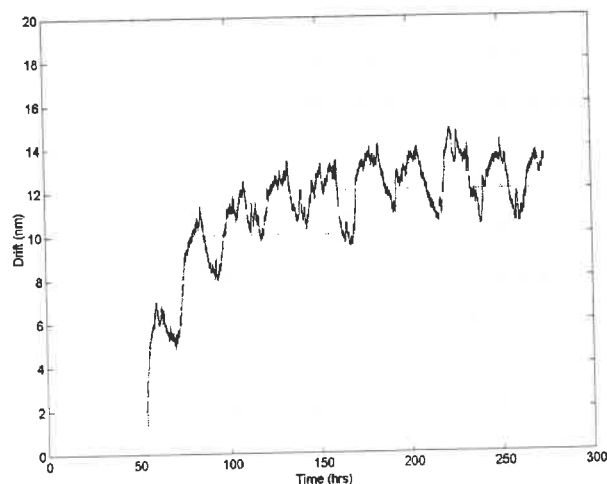


Figure 3: Interferometer Stability Results

Future Work

A small-scale optic mount sample has been designed to hold the optic in a configuration similar to the actual optic mount. A photo of the sample is shown in Figure 4. The mount is made from superinvar and the optic bonded to the mount is made from low

expansion two-phase glass ceramic. Superinvar has been chosen as the material for a significant portion of the structure of the metrology loop in order to reduce the temperature susceptibility of the experiment. The contribution of the superinvar to the stability of the interferometer has been budgeted based on previous stability analysis of superinvar [2]. In addition, the combination of low expansion glass ceramic and superinvar mimics the combination of materials in a typical epoxy joint. The bottom side of the optic is lapped to $\lambda/20$ flatness (peak-to-valley) and coated with a thin layer of aluminum. While the thickness of the epoxy bond line in the actual mount is approximately 75 μm , an epoxy bond line of 250 μm is used in the test samples to magnify the effects of creep for the purposes of experimentation. The samples are mounted such that the mirror surface of the optic is parallel to the mating surface of the mount to less than 15 microns. A pair of these samples will be used to evaluate the stability in the unloaded condition.

A second set of samples to which a load can be applied has also been designed. The loading mechanism is designed to load the epoxy joint in one arm in shear by pulling on it and in the other arm by pushing on it. This effectively doubles the sensitivity of the system.

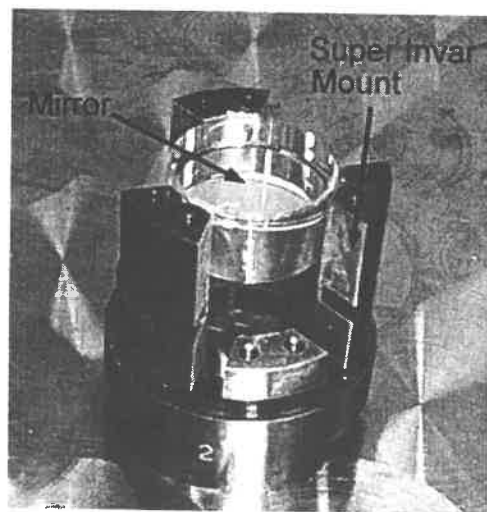


Figure 4: Epoxy Creep Samples

Conclusions

A balanced and temperature controlled plane mirror interferometric dilatometer has been constructed and used to qualify an interferometric measurement process at levels commensurate with its intended use for the measurement of the stability of epoxy joints.

Stability better than 0.4 nm/day has been demonstrated. The dilatometer is capable of measuring shear deformation of both loaded and unloaded epoxy joints

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NEW STRUCTURED AIR BEARINGS WITH COMPLEX THIN CHANNELS WITHOUT NOZZLES

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Air bearings are used in different applications for rotary and linear movements. Their main goal against the other kind of bearings is the very low friction due to the low viscosity of air. Therefore the load can be moved friction and stick-slip free. Air bearings are used for slides and spindles in high precision and ultraprecision machines, in coordinate measuring systems, etc.

Their characteristics are described by stiffness, carrying force, damping and air consumption. These characteristics are depending on the design of the air bearing and the air gap. Air bearings can be designed as aerostatic and aerodynamic bearings. Aerodynamic bearings are structured with spiral grooves for the needed aerodynamic effect. Conventional aerostatic bearings are mostly designed with a central nozzle in a chamber [1]. On the way to the ambient the supply air passes the nozzle, the chamber and the gap. Different types are realized with combination of nozzles, chambers and channels or only with porous sinter materials. Using nozzles limit the carrying force of the air bearing because of the reduce of supply pressure by the throttling loss at the nozzle. Additionally the high air volume in the chamber (dead volume) reduces the dynamic stiffness of the air bearing. Our development of the new design of air bearings aims the reduce of the dead volume and the use of the supply pressure without losses. The idea was to use an outlet hole without a nozzle and distribute the supply pressure on the surface of the bearing with channels. Additional channels with some μm depth shell used for the needed throttle for the bearing. For the design of the structuring we used a finite element calculation with a three-dimensional grid model of the air bearing structure. The needed fluidic equations (Navier-Stokes, Reynolds, etc.) were simplified with assumptions to the following equation:

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p^2}{\partial x} \right) + \frac{\partial}{\partial y} \left(h^3 \frac{\partial p^2}{\partial y} \right) = 0 \quad h = \text{local depth of channel, } p = \text{load pressure}$$

With the coordinates x, y and the depth of the channel h it is possible to describe the surface structuring of the air bearing. Different types of structured surfaces with combinations of linear and angular channels were modeled, [Figure 1](#). This types were selected to find out the influence of the depth h and the width of the channels as well as radial and circumference channels. We fixed the outer diameter, the center outlet hole, the limitation for the structuring area and the width of the channels for the first calculations ([Figure 1](#), type 6). The selected types of structured surfaces were described in an triangle mash, shown in [Figure 2](#) for the type 8, which contains the necessary information for the surface geometry. After the finite element calculation of the equation the distribution of the local load pressure depending on the air gap was calculated. The isobaric graph and the 3D-pressure distribution is shown for type 8 in [Figure 2](#). The following integration of the load pressure distribution results the carrying force. This calculation was done for air gaps between $1\mu\text{m}$ and $15\mu\text{m}$ for the types shown in [Figure 1](#). The stiffness was calculated by the results for the carrying force at different air gaps.

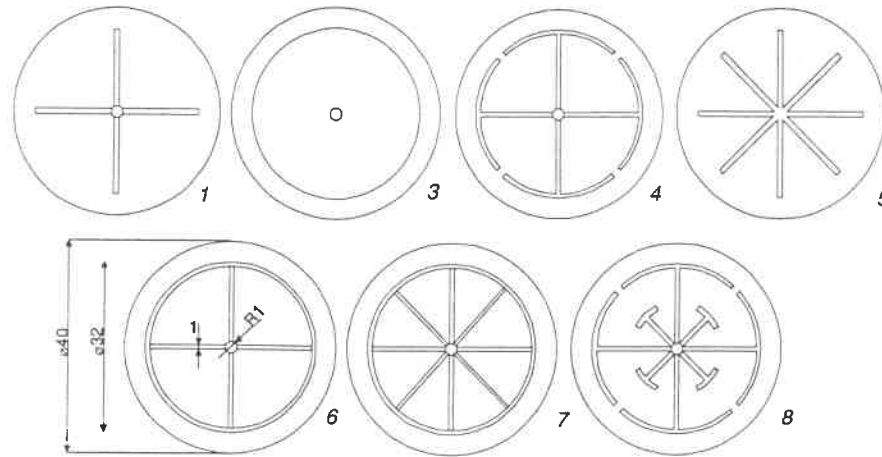


Figure 1: The chosen types of surface structures of air bearings for modeling and calculation

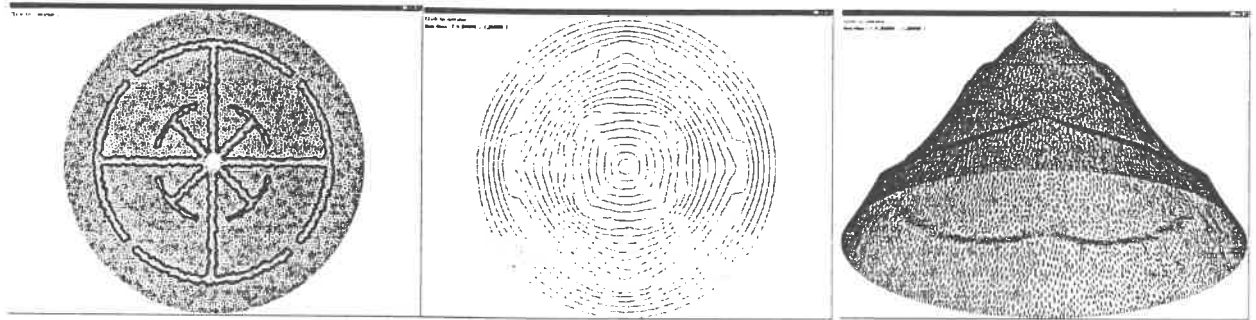


Figure 2: Triangle mesh of type 8 (left), calculated isobaric distribution (middle), calculated 3D load pressure distribution in the air gap (right), $h = 25\mu\text{m}$

In **Figure 3** the carrying force and the stiffness for all types are shown. The best stiffness was calculated for type 1 and 2. The best carrying force for type 7 and 8. Comparing both values the types 7 and 8 have the best values. Comparing the results we could determine that radial channels influence the stiffness and circumference channels influence the carrying force. A further interesting result was that the air gap at the maximum stiffness could be changed by the depth h of the channels. With higher h the air gap at the maximum stiffness rises also. Therefore we chose an h of $30\mu\text{m}$ to have the working area at an air gap of 5 to $6\mu\text{m}$. The calculations show also, that the achievable carrying force and stiffness is much higher compared to bearings with nozzles.

Our next step was to check the calculated with experimental results. Therefore we decided to manufacture the types 7 and 8 to compare the calculated with the experimental results. The main limitation in the finite element calculation was the mash, which does not allow finer or more complex structure types. Therefore we wanted to manufacture further types, where we could use the results of the calculations. The aim was to find different combinations of radial and circumference channels to optimize stiffness and carrying force.

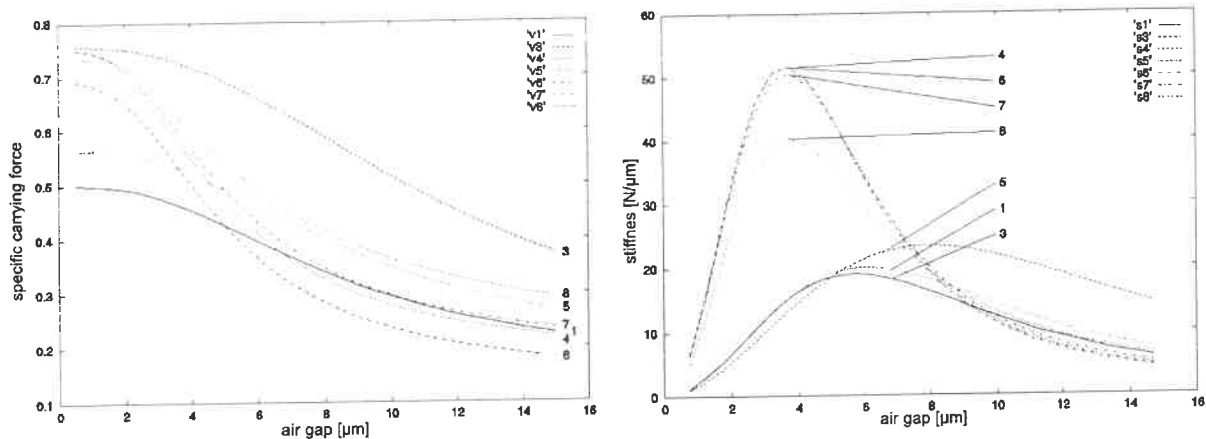


Figure 3: Calculation results for specific carrying force ($F/F_{\max}$) and stiffness of the chosen types with $h = 10 \mu\text{m}$, supply pressure 6 bar

Because of the desired high surface quality in form and roughness for air bearings we decided to manufacture the air bearings with a fast tool servo system on an ultraprecision lathe. The actuator of the fast tool servo system is a low voltage piezoelectric translator with a stroke up to $35 \mu\text{m}$, shown in Figure 4. The position of the diamond tool, which is guided by leaf springs, is measured by a capacitance sensor. The own build analog power amplifier allows a high bandwidth up to 1 kHz and a low system noise of 2 nm.

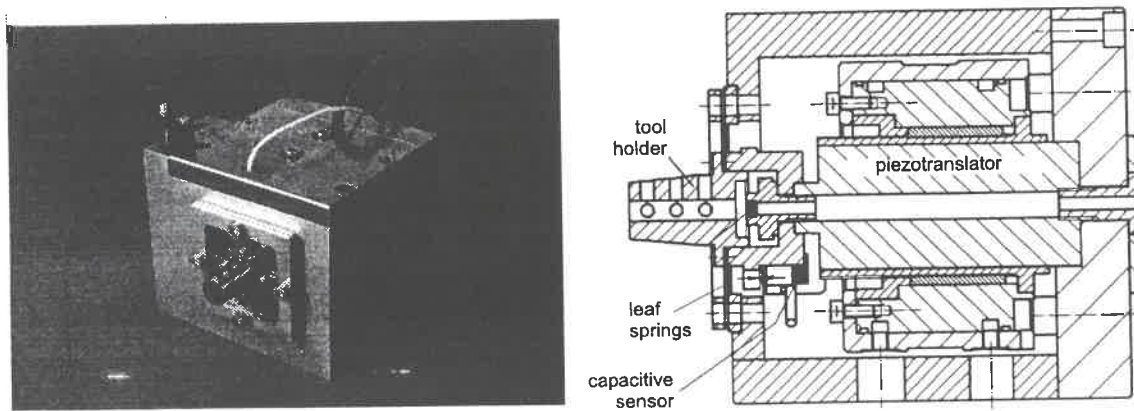


Figure 4: Piezoelectric fast tool servo system for diamond turning of non-rotational symmetric and structured surfaces

Figure 5 shows a picture of the manufactured and tested air bearings. In addition to type 7 and 8 we designed two further types we call „fountain“ and „plant“ which can not be modeled with the mesh of the finite element program. Next we tested the main characteristic of the new designed bearings.

The results in Figure 7 show that the characteristics of the type plant is excellent. Available values for air bearings with nozzles or porous sinter materials are in the area of 350 N carrying force and at $40 \text{ N}/\mu\text{m}$ stiffness at 6 bar supply pressure and an air gap of nearly $5 \mu\text{m}$. Values for

damping are not available. Our further work will concentrate testing further surface structuring and testing of the bearings in ultraprecision applications

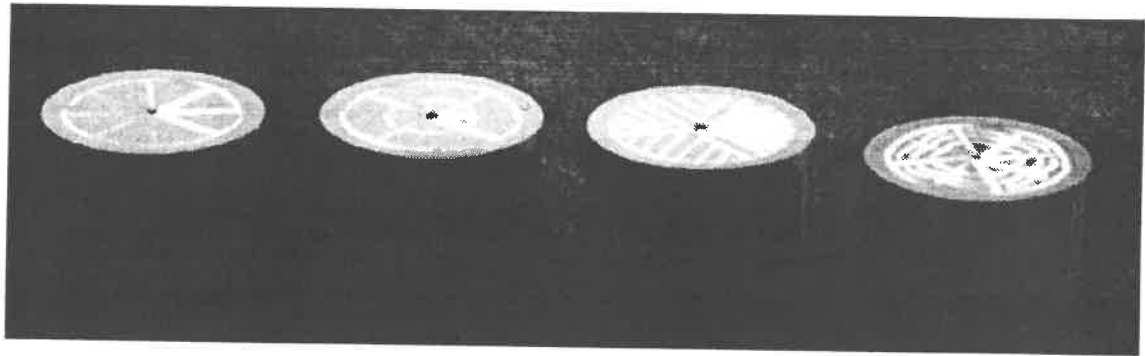


Figure 5: Picture of the manufactured air bearings with the fast tool servo system.

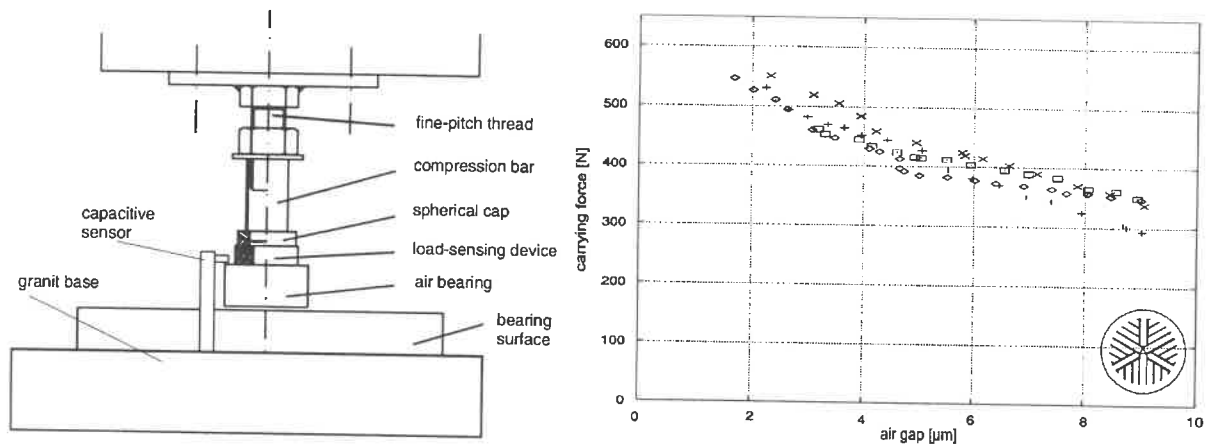


Figure 6: Measurement set up and measured carrying force results for the type „plant“

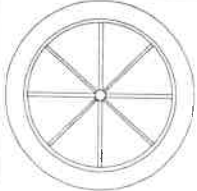
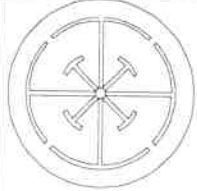
				
	type 7	type 8	fountain	plant
carrying force [N]	384	342	386	410
stiffness [N/ μm]	40	43.5	45	49
damping	0.045	0.072	0.054	0.049

Figure 7: Summarized characteristics for the four manufactured air bearings at an air gap of 5 μm , supply pressure 6 bar

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A Surface Current Density Model of Magnetic Suspension for Optical Pickup Heads

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I. Introduction

Magnetic repulsion and attraction are useful as a means for providing quiet, clean and frictionless suspension for future tracked ground transportation vehicles and for providing high-reliability, long-lived, friction free bearing for unique new devices and components[1].

Dual lens actuator for DVD/CD compatible pickup heads makes use of magnetic suspension mechanism for it can produce the non-contacting spring force, which enables the easy switching DVD position to CD position or vice versa. Although the suspension is an important mechanism in the dual lens actuator design, there have been few studies for systematic approach to design the mechanism. One way to design the mechanism is FEM (finite element method). However the approach needs a lot of time and cost. In this article, a guideway to design magnetic suspension is presented using surface current density model of uniformly magnetized material. Then the validity of the method is investigated by using FEM.

II. Dual lens actuator for DVD/CD compatible pickup heads

Recently, DVD has been introduced as a new optical data storage media, so that DVD/CD compatible driver is needed. Hence, a the dual lens actuator has been proposed, which the actuator can read DVD and CD using DVD and CD lens respectively[2]. The actuator has its appearance presented in Fig. 1. The axis enables the bobbin vertical and rotational motion only and suppresses parasitic tilt motions.

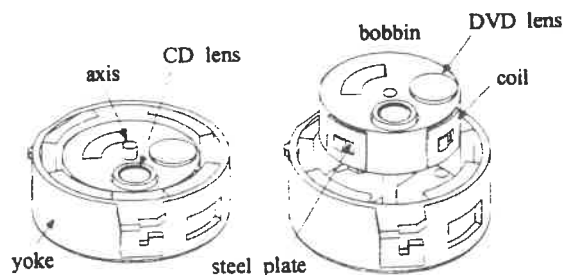


Fig. 1 Appearance of Dual Lens Actuator

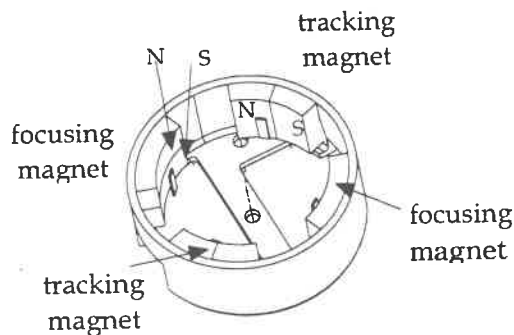


Fig. 2 Magnets, Yoke and Steel Plates

$$A(\mathbf{r}) = \frac{\mu_0}{4\pi} \int_{V_0} \frac{\nabla' \times \mathbf{M}}{|\mathbf{r} - \mathbf{r}'|} dv' + \frac{\mu_0}{4\pi} \int_{S_0} \frac{\mathbf{M} \times \mathbf{n}}{|\mathbf{r} - \mathbf{r}'|} da' \quad (2)$$

Using Eq. (1) and (2) we can define the magnetization current density of the as follows. $\mathbf{J}_M$ is volume current density [A/m²] and $\mathbf{j}_M$ is surface current density [A/m] [4].

$$\mathbf{J}_M = \nabla \times \mathbf{M} \quad (3)$$

$$\mathbf{j}_M = \mathbf{M} \times \mathbf{n} \quad (4)$$

The B vs. H and M vs. H hysteresis of a ideal permanent magnet is in Fig. 4. The operation point of the magnet is on the curve in the second quadrantal plane if the demagnetizing field is less than $M H_c$. In this case the magnetization is M_r . In uniformly magnetized rectangular prism the volume current is zero and the surface currents assume the shape of a solenoid of infinitesimal thickness that coincide with the nonpolar side surface of the magnet.[1]

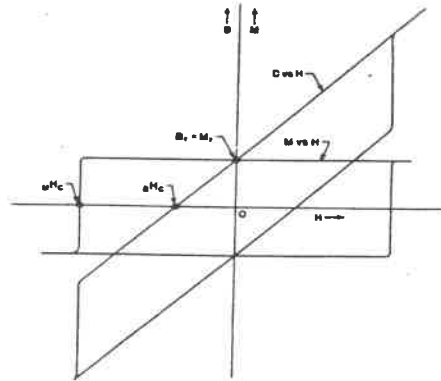


Fig. 4 Hysteresis of a ideal permanant magnets

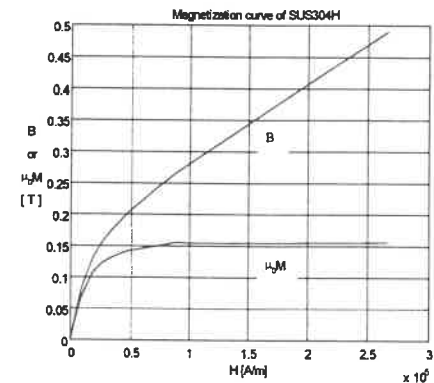


Fig. 5 Magnetization curve of SUS304H

Magnetization curve of the steel plate (SUS301H in JIS) is obtained by experiment in Fig.5. The magnetization of the steel plate will be saturated for the plate is very thin, the air gap between the magnets and the plate is very small, and the magnet has very high residual magnetic flux density. If the plate is saturated and the material is homogeneous and isotropic, the volume current is also zero and only the surface current is survived in nonpolar plane as the above.

Consequently, the uniformly magnetized material can be modeled as a conductive sheet which has the surface current density. The magnetic force caused by the conductive sheets can be obtained by Ampere's law of force [1].

$$\mathbf{F} = \frac{\mu_0}{4\pi} I I' \oint_C \oint_C \frac{\mathbf{r} (d\mathbf{l} \cdot d\mathbf{l}')}{r^3} \quad (5)$$

where $d\mathbf{l}$ and $d\mathbf{l}'$ are infinitesimal length elements along the paths of the currents I and I' , respectively, and $\mathbf{r}$ is the position vector between them.

V. Design result

The z-directional force between S_1 and S_1' can be obtained as follows. Note that the integrand in Eq. (5) is vanished if the two infinitesimal length elements are perpendicular.

$$F_{11'z} = -\frac{\mu_0}{4\pi} j_M j_m \int_{-t_M}^0 \int_{-h_M/2}^{h_M/2} \int_{-b_M/2}^{b_M/2} \int_{-b_m/2}^{b_m/2} \frac{(z' + \delta - h_M) dy' dy dz' dx}{[(g + t_m - x)^2 + (y' - y)^2 + (z' + \delta - h_M)^2]^{3/2}} \quad (6)$$

where g is air gap between the magnets and the plate, and δ is z-directional displacement of the plate. This forces must be calculated for all nonpolar planes and summed in order to obtain the F_z vs. δ relation as follows for design parameters in Table 1. (Note that the parameters with subscript 'M' is for the magnets and the parameters with subscript 'm' is for the plate.)

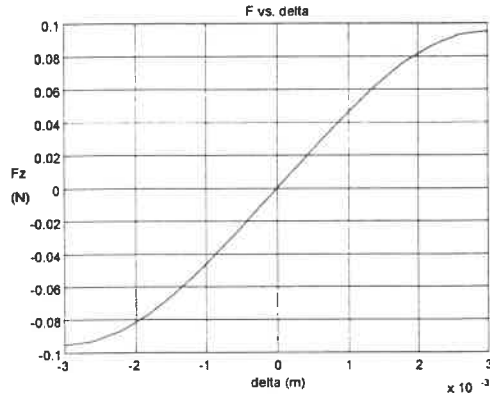


Fig. 6 z-directional force vs the displacement of the direction

magnets		steel plate	
b_M	10 mm	b_m	5 mm
h_M	10 mm	h_m	5 mm
t_M	2 mm	t_m	0.25 mm
j_M	1.2080e6 A/m	j_m	1.2354e5 A/m
g		0.5 mm	

Table 1 Design parameters of the magnetic suspension

VI. Conclusion

The stiffness of the suspension is approximately 45 ± 2.5 N/m in z-directional moving range, ± 1 mm. In the range the suspension is so linear that it can guarantees the linearity. Using finite element method, the very close stiffness can be obtained (43 N/m).

This approach enables us to predict the suspension force caused by the displacement along z-axis. So, the stiffness of the suspension can be calculated. The model presents the systematic way to design the magnetic suspension mechanism. The FEM simulation guarantees the validity of the modeling.

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The four steel plates are set 90° apart each other and facing to the focusing magnets and the tracking magnets. If the steel plates move, the magnetic force will be exerted to make the plates return to their equilibrium position[3].

III. Magnetic suspension mechanism

The mechanism has two permanent magnets and one steel plate, as presented in Fig. 1. The magnetization M_1 , M_2 and M_3 of each magnet and plate are depicted by solid arrows. The steel plate, SP can move along z -direction only. Two permanent magnets, PM_u and PM_l are fixed. The suspension is in equilibrium if the z -coordinate of SP center is zero. The origin, O is at the middle of the common edge of PM_u and PM_l . If SP moves along z -axis, the magnetic force makes the magnet return to its equilibrium.

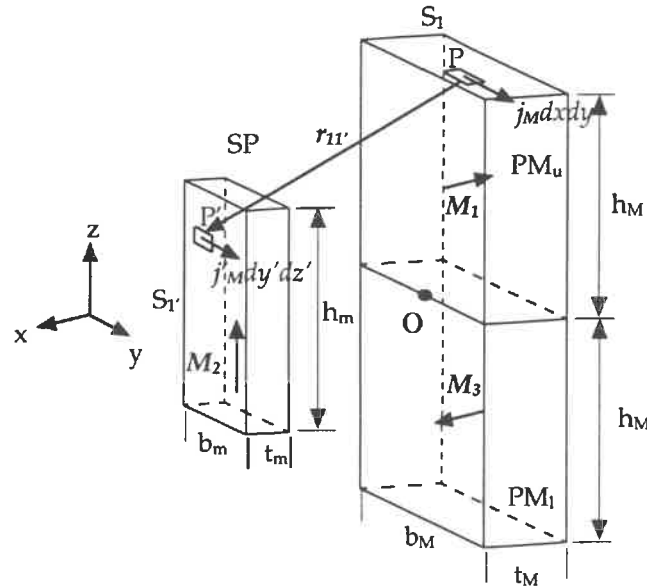


Fig. 3 Magnetic suspension mechanism

In this paper, the simple magnetic suspension shown in Fig. 3 is analyzed and designed. The magnetic suspension in the pickup heads (Fig. 3) can be also designed based on this approach.

IV. Surface current density model

The magnetic vector potential whose curl is magnetic flux density may be written as follows for a volume, V with a specified current distribution, $J(r')$.

$$A(r) = \frac{\mu_0}{4\pi} \int_V \frac{J(r')}{|r - r'|} dv' \quad (1)$$

The vector potential of magnetized material whose volume is V_0 and magnetization is M can be expressed as following equation.

EVALUATION OF THERMAL COMPENSATION MODELS ON A MACHINING CENTER

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Introduction

Machine tool positioning accuracy varies with the thermal state of the machine. As electrical energy is input into the servomotors, hydraulic pumps and other machine systems, energy dissipates into the part, atmosphere, and most importantly the machine's structure. This transfer of energy throughout the machine results in temperature changes and thus structural deformations that change the machine's accuracy. In order to mitigate these detrimental accuracy variations, models have been employed that try to predict and correct for these variations based on discrete temperature readings from around the machine [3,4,6].

While such models have shown adequate improvements in mitigating the thermal effects, industry has yet to adopt such an approach. Industry's reluctance is probably caused by the time intensive measurements, multitude of thermal sensors required, and uncertainty of the compensation's stability.

To address these issues, three thermal models were evaluated on a 3-axis machining center using the Laser Ball Bar (LBB) as the measurement instrument. The LBB permits data collection of all 21 error components simultaneously on a 3-axis machine. This reduces the measurement time to a single thermal cycle as opposed to the cycle per error component required with conventional equipment. To evaluate thermal sensor importance, a model requiring only four sensors was compared to two models utilizing thirty-one thermal sensors. Compensation stability was evaluated by monitoring a system over a nine month period.

The Models

The first thermal model utilizes thermal sensors placed on the machine's glass scales to compensate for their expansion and contraction. This type of model has been used by previous researchers and is commonly referred to as a first order thermal model [5,7]. In addition to the scale correction, another sensor is used to monitor the part temperature to further compensate the scales to behave like the part material at temperature, to insure correct dimensions at 20°C.

$$SCALE\ ERROR = \delta_X(X) + [(T_{scale} - T_{REF})\alpha_{scale} - (T_{part} - 20^\circ C)\alpha_{part}]X$$

$\delta_X(X)$ - Scale error measured at some temp, T_{REF} .

T_{scale} - Temperature of scale before correction.

T_{part} - Temperature of part before correction.

α_{scale} - Coefficient of expansion for the scale.

α_{part} - Coefficient of expansion for the part.

This model was selected for comparison because it is completely deterministic and uses very few thermal sensors. It also serves as a good reference point for models using many more sensors. The remaining parametric errors are compensated based on measurements taken while the machine is in a cold state. Homogeneous Transformation Matrices (HTM) were used to derive the tool point error equations from the parametric errors.

$$\begin{aligned}
P_X &= \delta_X(X) + \delta_X(Y) + \delta_X(Z) - Y\{\epsilon_Z(X) + \epsilon_Z(Z)\} - Y_T\{\epsilon_Z(X) + \epsilon_Z(Y) + \epsilon_Z(Z)\} \\
&\quad + Z_T\{\epsilon_Y(X) + \epsilon_Y(Y) + \epsilon_Y(Z)\} \\
P_Y &= \delta_Y(X) + \delta_Y(Y) + \delta_Y(Z) + X\epsilon_Z(Z) + X_T\{\epsilon_Z(X) + \epsilon_Z(Z)\} \\
&\quad - Z_T\{\epsilon_X(X) + \epsilon_X(Y) + \epsilon_X(Z)\} \\
P_Z &= \delta_Z(X) + \delta_Z(Y) + \delta_Z(Z) - X\epsilon_Y(Z) - X_T\{\epsilon_Y(X) + \epsilon_Y(Y) + \epsilon_Y(Z)\} \\
&\quad + Y\{\epsilon_X(X) + \epsilon_X(Z)\} + Y_T\{\epsilon_X(X) + \epsilon_X(Y) + \epsilon_X(Z)\}
\end{aligned}$$

(P_X, P_Y, P_Z) - Tool point positioning error.
 (X_T, Y_T, Z_T) - Tool offset from measurement tool position.

The measured parametric errors were fit with fourth order polynomials and the coefficients input into the model in the controller.

The remaining two thermal models were based on an Adaptive Resonance Theory (ART) neural network using fuzzy logic [2,6]. These models utilize the same tool point error equations derived for the first order model, but handle the thermally induced variation of all twenty-one parametric errors with the neural network. Seven fuzzy ART map networks were utilized to predict the parametric errors from thirty-one 'T' type thermocouples mounted to the machine structure, Figure 1.

The two neural network models' architecture were identical, only their training cycles differed. One network was trained with thermal cycling consisting of axis and spindle operation without machining. The other network was trained with thermal cycling composed of metal removal requiring different amounts of power consumption. This was done to evaluate what effects actual machining might contribute to the machine's thermal deformation as opposed to the traditional method of cycling without machining. Previous research has been limited to thermal cycling without machining due to the delicacy of the measurement instruments or their inability to be removed from the work zone without loss of reference. The LBB overcomes these limitations because it can be removed from the work zone during machining, while its magnetic sockets remain in place retaining reference.

Training Cycles and Measurement Procedures

The first order thermal model did not require any thermal cycling since its thermal prediction is purely analytical. Measurements for the 21 parametric errors were taken while the machine was in its power-on cold state. This took approximately 20 minutes. The measurement set-up is shown in Figure 2.

The neural network models required training with the machine in as many thermal states as possible to accurately predict the parametric error variation with temperature. For the non-machining thermal cycle, two basic duty cycles were chosen for training. The first consisted of actuating the machines axes at 5 m/min while the spindle was operating at 2500 RPM. The parametric errors were measured in the cold state, followed by warming at these speeds for approximately 30 minutes, followed by measurements of the parametric errors. This approximate one hour cycle was repeated over an eight hour period at which time the machine was allowed to cool while measurements were collected every hour for four hours. This provided measurement data at twelve thermal states. The second duty cycle, conducted on a different day, was identical except for feeds and speeds of 10 m/min and 5000 RPM, resulting in twenty-four total thermal states for the first network model.

For the network trained with machining, the cycling was similar to the non-machining except the load free actuation was replaced with machining of aluminum requiring 1 kW of power on one day and 5 kW on another.

The 20 minute measurement procedure consisted of visiting 10 equally spaced points bidirectionally along each of the three axis, lengths of (603,504,405)mm respectively for the X, Y, and Z axes. At each measurement cycle the LBB was reinitialized and the three base lengths remeasured. This was followed by six sequential measurements between the base sockets and tool sockets as the CNC measurement program was repeatedly executed for each of these six socket measurements, Figure 3.

Evaluation Tests

To evaluate the models, diagonal measurements were made inside the compensation zone using the LBB, Figure 4. Also part machining tests were conducted using the B5.54 precision positioning test part [1]. The diagonal

tests were taken at four different thermal states achieved with axis actuation different than used during training. Similarly, the machining tests were conducted at four different thermal states achieved with a combination of axis actuation and scrap machining different from training. The stability tests were conducted by using the first order thermal model built on 8-20-97 and measuring the translational parametric errors over a nine month period with this compensation model. The results of the diagonal tests are shown in Table 1, and compare the total range of error between the models over the four thermal states

TABLE 1

Diagonal Error Ranges over 4 Thermal States [microns][improve ratio]								
	No Comp		1st Therm		ANN #1		ANN #2	
Diag #1	33.5	-	5.8	5.8	12.8	2.6	11.8	2.8
Diag #2	34.0	-	8.3	4.1	12.1	2.8	14.7	2.3
Diag #3	19.8	-	8.6	1.7	11.3	1.8	11.7	1.7
Diag #4	23.6	-	12.1	1.9	8.3	2.8	8.7	2.7

Similar results were found for the machining tests, with improvement ratios of between about 2-3X. As in the diagonal tests the 1st order model performed as well or better than the neural networks. The stability test results are shown in Table 2 and indicate the compensation is still beneficial after nine months. During this period the machine's history and use consisted of the warm-up and part evaluation machining and a spindle replacement between the 12-19-97 and 5-19-98 tests.

TABLE 2

Parametric Error Ranges During Stability Tests						
	No Comp	8-20-97	9-18-97	10-14-97	12-19-97	5-19-98
XY (arc-sec)	7.5	-0.7	-0.1	0.4	2.2	-0.7
XZ (arc-sec)	-10.9	1.7	2.9	1.7	3.6	6.5
YZ (arc-sec)	12.6	1.1	0.2	0.1	3.3	5.8
$\delta x(X)$ (μm)	11.9	7.1	7.0	4.9	7.7	10.6
$\delta y(Y)$ (μm)	9.7	6.1	5.7	7.5	4.6	11.0
$\delta z(Z)$ (μm)	11.0	4.0	3.1	3.2	3.7	2.8
$\delta y(X)$ (μm)	25.3	1.9	2.1	2.0	4.1	5.6
$\delta z(X)$ (μm)	8.8	3.1	4.3	3.7	2.5	4.7
$\delta x(Y)$ (μm)	8.5	1.8	2.5	3.1	3.3	2.5
$\delta z(Y)$ (μm)	3.7	1.4	1.4	1.3	2.7	3.4
$\delta x(Z)$ (μm)	3.0	4.4	3.8	3.6	3.8	4.4
$\delta y(Z)$ (μm)	2.4	2.1	1.5	1.7	2.0	1.5

Conclusions

A simple first order thermal model that can be parameterized in less than one hour was found to be equal or superior to the more elaborate neural network models. Stability tests indicate that such a model might need

retraining about every six months. The data suggests that the capability now exists to significantly reduce thermal error with very little down time required for training and with a small addition of sensors.

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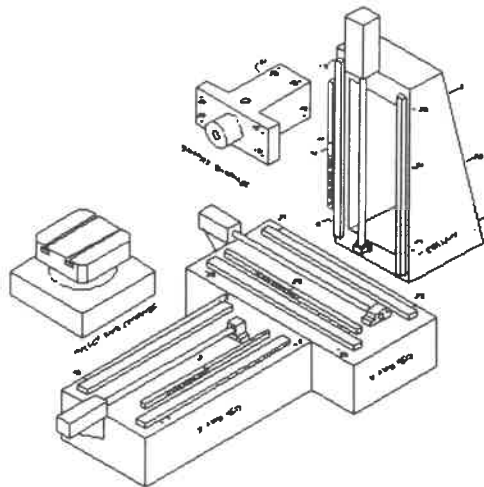


Figure 1 Thermocouple locations.

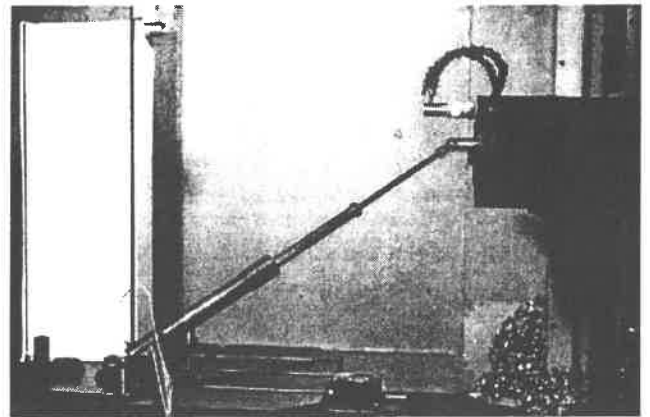


Figure 2 Measurement set-up.

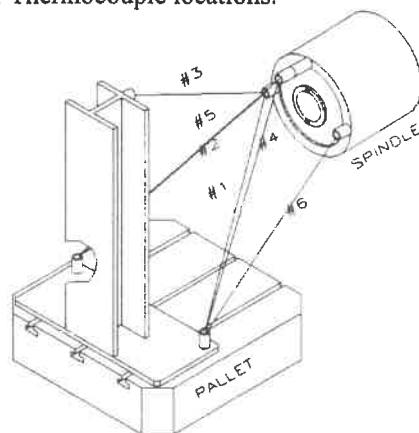


Figure 3 Six measurement legs.

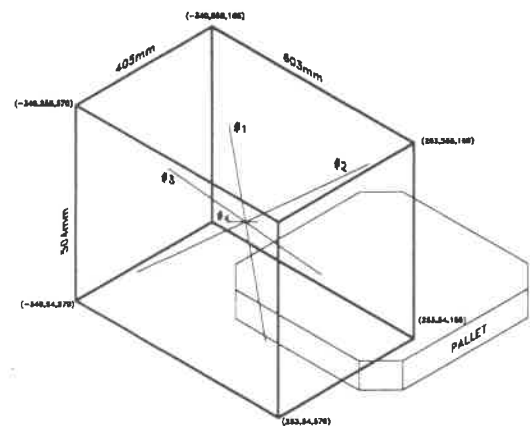


Figure 4 Diagonal measurements in compensation zone.

Experimental Analysis of Friction and Wear Characteristics of Ceramics Under No-lubrication Plane-Plane Contact Conditions

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1. Introduction

Cast iron and steel have been used mainly as structural materials for machine tools. Recently, however, a variety of new structural materials have just come into use in the development of high-performance (ultra precision or ultra-high speed) machine tools. Various new ceramics, fiber-reinforced plastics (FRP), and concretes have been put into practical in the development of machine tools. Of these new materials, ceramics are superior to cast iron and other metals in terms of strength, specific Young's modulus, and resistance to heat, wear, and corrosion. In addition, ceramics also have the advantage of a very low susceptibility to deterioration by aging. Given these excellent properties, ceramics have been partially adapted for use in the main spindles and slideways of machining centers¹⁾. One form of this material, alumina ceramic, is an excellent structural material for machine tools which offers the general characteristics of ceramics, as well as the following three advantages: (1)The ability to fabricate large-size machine elements. (2)Availability in a range of inexpensive yet-stable quality materials. (3) High machinability. Though silicon carbide is more expensive and less machinable than alumina ceramic, it has a superior sliding characteristic and is expected to be adopted as another of the main structural materials. In the present study, we developed a plane-plane contact-type high-speed friction tester, a new experimental device to address the recent tendency toward high speed, and we experimentally analyzed the friction and wear characteristics of several ceramics. We summarize our results below.

2. Experimental Device and Specimens

Fig.1 shows an outline of the experimental device we developed. The carriage is supported by a maintenance-free slideway structure. Reciprocating motions are transmitted and transformed into rotational motions through the motor with speed-change gears and the pulley to the crank shown in the figure. This crank and the carriage are coupled with each other by the connection rod, and together produce sine wave motions. The average sliding speed (hereinafter called the sliding speed) is adjustable between 1.95m/min and 34.3m/min. The fixed specimen mounted on the specimen holder sits on the device body, while the movable specimen is fixed on the carriage. The specimen contact surface is loaded by a spring, and load is measured by a strain-gauge-type load transducer(F2). Frictional force is measured by a strain-gauge-type load transducer(F1) placed between the fixed specimen holder and the device body. As the coefficient of friction, we used the average of the values measured by the load transducers F1 and F2, obtained in a 62mm area of the 65mm stroke (excluding both end portions) by drawing signals from the load transducers into a personal computer through a simultaneous sample hold sensor interface board (containing a conversion power supply, an amplifier, and an A/D converter). The size of the fixed specimen was 30mm x 50mm, and its apparent area was 1,500mm². In consideration of the problems of the global environment recently drawing widespread attention, the test specimens were placed under a no-lubrication condition, washed in an ultrasonic cleaner, and dried completely. All experiments were conducted in a temperature controlled room kept at a temperature of $20 \pm 0.5^{\circ}\text{C}$ and a humidity of $50 \pm 5\%$. The specifications of the specimens were

as follows : Alumina ceramics (99.5% Al_2O_3): bulk density= 3.8g/cm^3 ; bending strength= 324MPa ; coefficient of linear expansion= $6.8 \times 10^{-6} / ^\circ\text{C}$; thermal conductivity= $5\text{W/(m}\cdot\text{k)}$ (all catalog values): ground. Silicon carbide(Sic): bulk density= 2.75g/cm^3 ; bending strength= 557MPa ; coefficient of linear expansion= $4.3 \times 10^{-6} / ^\circ\text{C}$; thermal conductivity= $46\text{ W/(m}\cdot\text{k)}$ (all catalog values): ground. Two types of ceramics were used, as shown above.

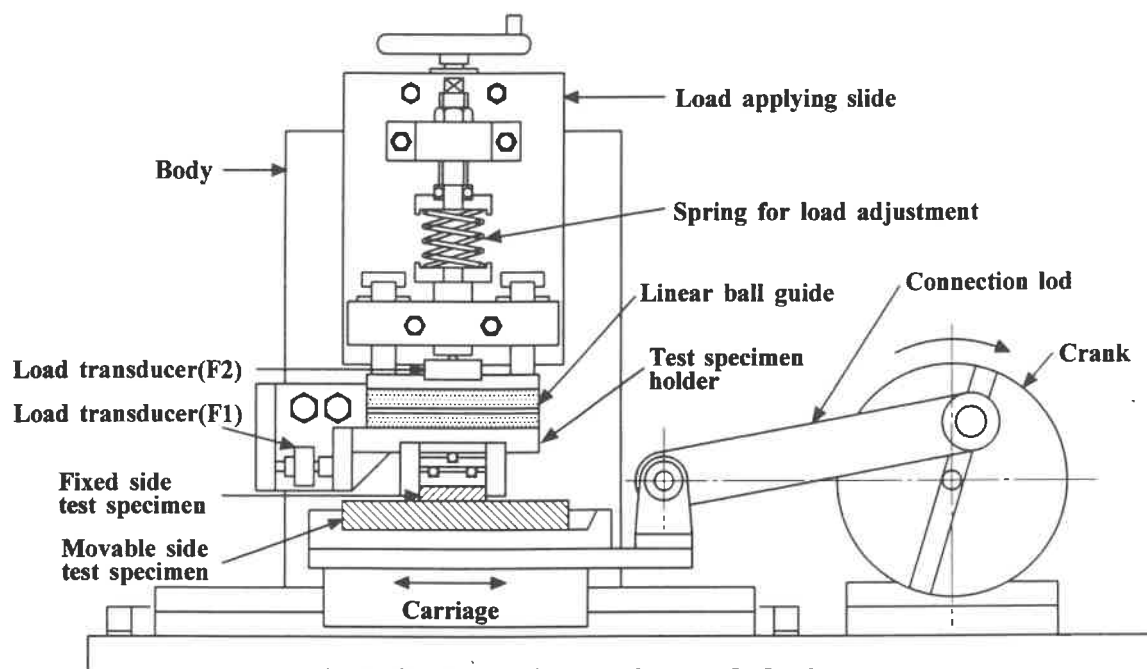


Fig.1 Outline of experimental device

3. Results of Experiments

3.1 Combination of alumina ceramic - alumina ceramic

Fig.2 shows the relationship between the sliding distance and the change in the coefficient of friction and the surface roughness (Ra) in the combination of alumina ceramic - alumina ceramic. The experiments were performed under the conditions of contact pressure $P=0.15\text{MPa}$ and sliding speed $V=1.95\text{m/min}$. The result indicates that the coefficient of friction increased significantly from the start of the first experiment to the sliding distance of 200 - 300m. However, as the sliding distance increased past these values, the coefficient of friction remained unchanged. The surface roughness decreased sharply just after the start of the first experiment, but the rate of decrease tended to be very gradual with further increases in the sliding distance. Additionally, we discovered a correlation between the surface roughness of the specimens and the coefficient of friction: when the surface roughness of the specimens became stable, the coefficient of friction also stabilized. Fig.3 shows the relationship between the contact pressure and the coefficient of friction in the combination of alumina ceramic - alumina ceramic obtained in experiments conducted at a sliding speed of $V=1.95\text{ m/min}$. and contact pressure of $P=0.05\text{MPa} - 0.35\text{MPa}$. This range of contact pressure was selected in

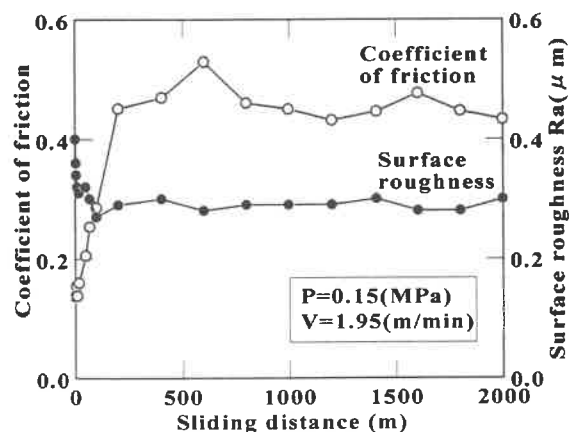


Fig.2 Relation between sliding distance and coefficient of friction and surface roughness (alumina - alumina)

consideration of the allowable pressures on the steel slideways of machine tools: approximately 0.098MPa in a normal condition, 0.34MPa in a well-lubricated condition, and approximately 0.029MPa when precise sliding motions are required²⁾. The total sliding distance per measured data was 500m. In the first experiment, the coefficient of friction was measured by increasing the contact pressure gradually from 0.05MPa to 0.35MPa. In the second and subsequent experiments, the specimen used in the first experiment was washed in an ultrasonic washer, dried completely, set back in place, and tested once more. The first experiment showed that the coefficient of friction increased almost linearly as the contact pressure increased, but it varied widely between 0.3 and 0.42. The inclination of the line became gentler as the experiments were repeated. The increase of the coefficient of friction (0.35 - 0.38) caused by the increase of the contact pressure in the fifth experiment was much smaller than that in the first experiment, probably because of the phenomenon of fitting. The surface hardness of the fixed specimen (Ra) decreased from 0.78 μ m before the start of the first experiment to 0.57 μ m after the fifth experiment. From the above results, the surface roughness of the specimens was shown to be improved and the apparent contact area also grew larger. The coefficient of friction may have been less susceptible to the contact pressure and the sliding distance.

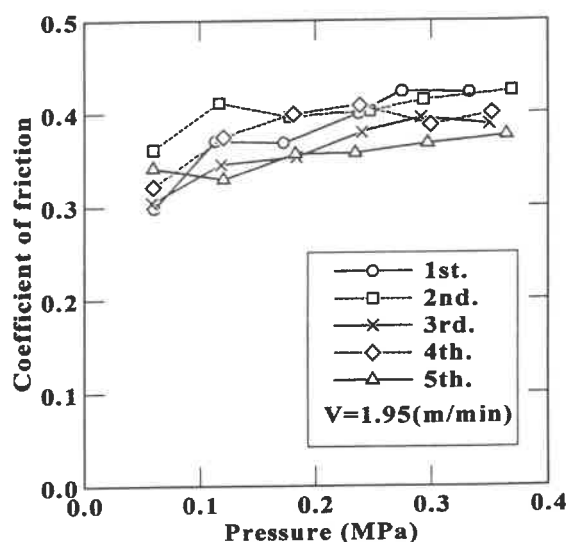


Fig.3 Relation between pressure and coefficient of friction (alumina - alumina)

3.2 Combination of silicon carbide - silicon carbide

Fig.4 shows the relationship between the contact pressure and the coefficient of friction in the combination of silicon carbide - silicon carbide. The experiment conditions for alumina ceramic - alumina ceramic were also applied to this combination. The number of experiments was reduced since only minimal changes were observed between the third and subsequent experiments. The coefficient of friction in the first experiment was 0.2 under the contact pressure of 0.05MPa and approximately 0.26 under the contact pressure of 0.36MPa. Although the coefficient of friction to the contact pressure in the third experiment slightly increased from that in the second experiment, the increase of the coefficient of friction with the increase of the contact pressure in the third experiment was smaller than that in the first experiment and stabilized. The surface roughness improved from 0.1 μ m before the start of the first experiment to 0.08 μ m after the last experiment. The rate of frictional change of the silicon carbide - silicon carbide combination was smaller than that of the alumina ceramic - alumina ceramic combination.

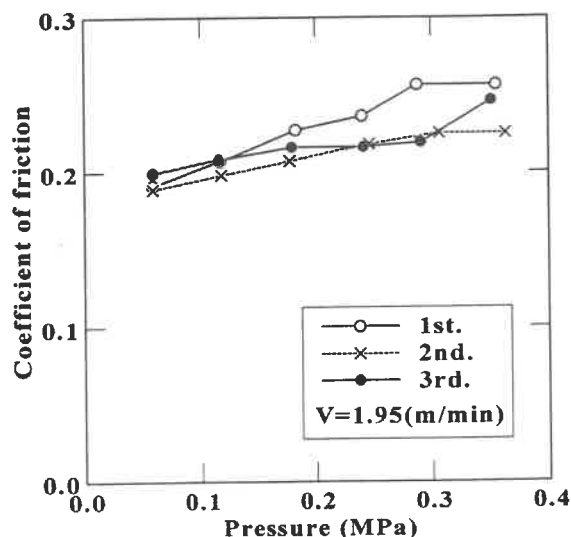


Fig.4 Relation between pressure and coefficient of friction (silicon - silicon)

3.3 Sliding velocity and coefficient of friction

Fig.5 shows the relationship between the sliding speed and the coefficient of friction in the alumina ceramic - alumina ceramic and silicon carbide silicon carbide combinations. All of the specimens used were the same as those used for the contact pressure and coefficient of friction experiments. The experiments were conducted under the conditions of contact pressure $P=0.2\text{MPa}$ and sliding speed $V=0.2\text{m/min} - 16\text{m/min}$. In the case of the combination of alumina ceramic - alumina ceramic, the coefficient of friction increased as the speed increased, proving that it was speed - dependent. Compared with the alumina ceramic - alumina ceramic, the silicon carbide - silicon carbide showed less speed dependency, and its coefficient of friction was as low as 0.18. Silicon carbide is thought to be more appropriate than alumina ceramic as a material for sliding parts.

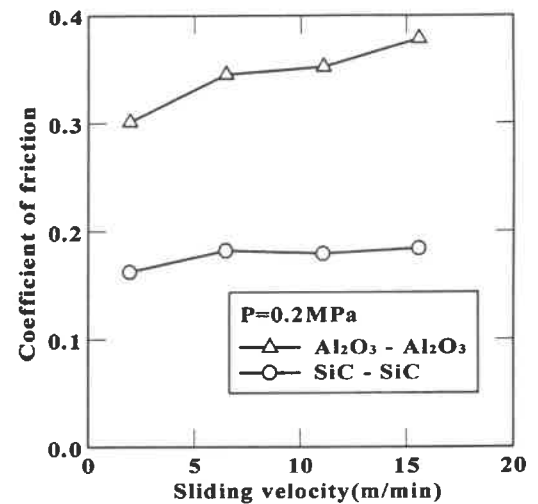


Fig.5 Relation between sliding velocity and coefficient of friction

4. Examination and Conclusion

Materials for sliding parts for machine tools must ensure excellent motion stability and positioning repeatability. Thus, to apply a ceramic as a material for sliding parts for machine tools, the coefficient of friction of the ceramic must remain stable against the sliding distance, the contact pressure, and the sliding speed, and it must be lower than the coefficient of friction of dry metals, i.e., 0.35 - 0.40 in the air³⁾. Here, we examined the characteristics of two types of materials subjected to high-speed sliding experiments to investigate the above two factors. Alumina ceramic shows a relatively high degree of coefficient of friction stability after the initial fitting, but it is not far superior to metal in terms of its coefficient of friction (0.38), which is almost equal to the dry coefficient of friction of metal. Silicon carbide, on the other hand, is a suitable material for sliding parts because of its low coefficient of friction (0.2) and in susceptibility to changes in operating conditions. In this analysis, we evaluated the friction and wear characteristics of combinations of materials of the same type. In the future, we will conduct further experiments using combinations of different types of materials and new materials, and we will provide basic design guidelines for applying new materials for use in sliding parts of machine tools.

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Sealing Effects on Leakage Flow of Oil Mist Lubrication for High Speed Spindles

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1. Introduction

Sealing plays important roles to have enhanced lubrications for high-speed spindles, which employ oil jet or oil mist lubrication system. High-speed spindles require non-contact types of sealing mechanism that prevent leakage of oil jet or oil mist. It is required that such seals should be designed to prevent leakage of oil jet or oil mist fluids. The design concept for contact type seals is concerned with the durability of the basic material or surface roughness. However, the design concept for non-contact type seals is concerned with geometric optimization to minimize leakages.

Gas or liquid has been used as a working fluid for most of non-contact type seals including the labyrinth seal. Previous researches mainly use single-phase (i.e., gas or liquid phase) flow analysis because the working fluid of ordinary non-contact type seals is mainly single phase. However, it is more reasonable to apply two-phases flow because high speed spindles use oil mist or oil jet lubrications. In this consideration, two-phase flow analysis must be applied to represent oil-air leakage characteristics.

Fig. 1 shows a conventional type of labyrinth seal (It is classified as "Cavity Flow Type" in this paper) and its analysis zone. This study categorizes geometry of mostly used non-contact type seals and analyzes each leakage characteristics to minimize leakages on the sealing area. Both of the turbulence and

the compressible flow model are also introduced in CFD (Computational Fluid Dynamics) analysis. As a results of CFD analysis in various cases, a protective collar type and an air jet type seal shows excellent sealing effects for oil mist lubrication. By the basis of analytical result, an adapted model is introduced to improve sealing capability of conventional non-contact type seals. It has a combined geometry of a protective collar type and an air jet type. Effect of sealing improvement is explained as decreasing of leakage clearance by air jetting. In this study, both of numerical analysis by CFD and experimental measurements are carried out to verify sealing improvements. Sealing effect of the leakage clearance and air jet magnitude are studied for various parameters in experiment. This procedure offers a methodological way of an enhanced seal design for high-speed spindles.

2. Leakage Flow Analysis

Oil jet or oil mist lubrication includes much of oil droplet that leads different flow characteristics from gas or air. Two-phase flow means each phase has different physical properties (i.e., velocity, temperature). To analyze a free surface flow, an oil-air two-phase analysis uses the Eulerian-Eulerian model to regard a divided space by each phase. Therefore, each phase has its own volume, velocity and temperature separately but shares a common pressure. Each phase can transport

such momentum and enthalpy within the flow.

In the analysis of leakage flows, turbulence and compressible flow model are also introduced. This investigation adapts the two-equation $k-\varepsilon$ model of Harlow-Nakayama. To solve compressible flow problems, the continuity and momentum equations must be supplemented by a transport equation representing the conservation of energy

A commercially available CFD code – PHOENICS (version 2.1) was used to analysis a flow field. In this analysis, a parametrical approach is carried on to obtain effects of geometric parameters. Pressure distribution, turbulence intensity and velocity profile are obtained to investigate leakage characteristics. Sealing capability is estimated by the amount of pressure drop between inlet and outlet assuming constant leakage flows. The inlet temperature is assumed to be 100 °C for high temperature generation inside of bearing, the environment temperature is assumed to be 20 °C, and the heat transfer is assumed to be adiabatic condition.

Effect of Design Parameters on Cavity Flow Type

Fig. 2 shows design parameters of Cavity Flow Type. The design parameters in Cavity Flow Type are depth and width of cavity on the same sealing space. Design optimization has been introduced to these parameters to minimize leakage flows.

Effect of Cavity Depth

Fig. 3 shows effects of cavity depth and width on pressure drop for non-dimensional form. The more pressure drop means the better sealing capability assuming constant leakage flow condition for each design parameters. Here, the U_i means an inlet velocity of leakage flow, the ρ means density of oil mist. When the W_0/C equals to 2.5 and 5.0, maximum pressure drop occurs at $H/C=2.7$. When the W_0/C equals to 10.0 and 15.0, maximum pressure drop occurs at

$H/C=1.3$. It is generally expected that deeper cavity makes bigger pressure drop. However, this result shows a possibility of lesser pressure drop in spite of deeper cavity. The pressure drops are decreased when the cavity depth exceeds this critical point ($H/C=2.7$ or $H/C=1.3$). Fig. 4 shows dead circulation flows in the cavity. Inflow to cavity is increased along with increase of cavity depth until the circulation flow does not occur in the cavity. When the cavity depth is increased over the critical point, circulation flow occurs in the cavity and it prevents inflow to the cavity. Therefore the critical point can be assumed to be a maximum depth which does not occur circulation flow in the cavity. The more inflow makes the more pressure drop to prevent leakage flows.

Effect of Cavity Width

Fig. 3 shows the wider cavity width makes the more pressure drop. Velocity profile in the cavity shows that the flow from narrow cavity width does not expand leakage flow volume enough to make large pressure drop. Pressure distribution results show most of pressure drops occur in the change of flow field from wide to narrow. Therefore, narrow cavity has poor capability to reduce leakage flow. For the purpose of design optimization, wider cavity is preferred than narrow one to improve sealing performance.

Effect of Design Parameters on Protective Collar Type

Fig. 5 shows design parameters of protective collar type. This type can be expected two kind of sealing mechanisms. One is changing flow direction into the cavity to increase turbulence. The other is isentropic process effect generated in minimum clearance at C1, C2 or C3. Therefore, design parameters of protective collar type are length and location of the collar. The length and location of minimum clearance can be determined by C1 and C2

Effect of Collar Face and End Clearance on Narrow Passage

Fig. 6 shows a comparison of pressure drop results on narrow passage ($H/W_0=0.1$) varying the face and end clearance (C_1 and C_2). Because of forced inflows to cavity, Protective Collar Type generates bigger pressure drop than Cavity Flow Type. However, the results show that isentropic effect is not so remarkable. Because of lack of flow expandable area before minimum clearance, the isentropic effect does not occur greatly. The collar position is preferred at front and end of cavity. Middle position generates lower pressure drop than front or end position. Effect of C_2 clearance shows the smaller clearance makes the larger pressure drop. As C_2 clearance become large, the isentropic effect is reduced and collar length is no more affects to pressure drop.

Effect of Collar Face and End Clearance on Wide Passage

Fig. 7 shows an effect of collar position (C_1 , C_2) on wide passage ($H/W_0=0.4$). The results show the isentropic effect on pressure drop evidently. Pressure drop differences are large as the C_1 clearance become wide. In case of wide passage condition, most of pressure drop occurs in C_1 clearance by isentropic effect. Therefore, minimum clearance should be located between the cavity front and the protective collar for effective sealing.

Effect of Design Parameters on Air Jet Type

Fig. 8 shows a design parameters of Air Jet Type. U_i , V_j and θ means inlet velocity, air jet velocity and air jet angle respectively. Design parameters are V_j , θ and cavity installation.

Effect of Jet Angle

Fig. 9 shows pressure drop results on air jet angle. Maximum pressure drop occurs at

90° jet angle. If the air jet angle exceed over 90° , its direction accelerates leakage flow and pressure drop decreased. Effective clearance for leakage flow is reduced by jetting amount. When air jet injects to 90° , the effective clearance become minimizes.

Effect of Air Jet Magnitude

Fig. 10 shows an effect of air jet magnitude on pressure drop with and without cavity. The results show the more jet magnitude makes the more pressure drop. Because the jet magnitude can not be increased illimitably, this type can be effectively used to sealing elements to apply low leakage flow.

Experiments of Adapted Seal Model

By the basis of analytical results, an adapted model is introduced to improve sealing capabilities of conventional non-contact type seals. Fig. 11 shows a schematic geometry of an adapted seal model. It has a combined geometry of a protective collar type and an air jet type seal. A protective collar, which makes a minimum clearance between the collar and cavity wall, is located at the inlet side of the cavity. Then air is injected against leakage flows in the minimum clearance. It makes an enhanced sealing performance along with increasing pressure drops. A clearance reduction effect can be expected by air jetting in this model although the clearance can not be physically close together. When the air is injected into the minimum clearance, the jetting air reduces an effective leakage clearance. This model can achieve excellent sealing performance although a rotating spindle and cavity have wide leakage clearance unavoidably.

An experimental device is constructed to test sealing performances of the adapted model. This device has oil mist generating nozzles to make oil mist lubrication environments. Sixteen air jet nozzles are installed to the circumferential direction for jetting air.

Experimental Results

Fig.12 shows an effect of jetting pressure on constant leakage pressures (from 0.0 to 0.2 Pa). Leakage pressures are proportional to the leakage flow at the 0.5mm clearance. The result shows that increase of jetting pressures increase total pressure drops. Thus, sealing performances can be improved by the air jet. This is very an effective way of the sealing enhancement in oil mist lubrication system for high-speed spindles.

Fig. 13 shows effects of leakage clearances on constant jetting pressures. As leakage clearances become larger, pressure drops decrease, which representing poor sealing performances with larger clearance like ordinary non-contact type seals. However, this figure shows inverse behaviors at the clearance between 0.25mm and 0.5mm. As diameters of jetting nozzles are 1.0 mm, leakage clearance of 0.25mm is not enough to adapt the air jet. However, in the case of 0.5mm leakage clearance, effects of air jets become dominant to generate pressure drops. Thus, the air jet itself can achieve good sealing performances although a spindle lubrication system has large leakage clearance.

Conclusions

Design parameters have been studied to minimize leakage in limited spaces, and a methodological study on geometrical optimization has been conducted. As results of the CFD analysis in various geometric cases are summarized as follows.

- (1) In case of Cavity Flow Type, cavity depth is optimized to produce maximum pressure drop.
- (2) Wide cavity width is preferred to reserve a flow expandable area.
- (3) In case of Protective Collar Type, large pressure drop from isentropic process can be achieved when the leakage passage is wide.
- (4) Minimum clearance should be located

between the wall on the flow inlet and the collar

- (5) In case of Air Jet Type, jet magnitude should be large compare to leakage flow.
- (6) 90° injection of jet angle produces best pressure drop.

By the basis of analytical results, an adapted model is introduced to improve sealing capabilities. Their sealing characteristics are investigated through experiments using the performance testing device. Effects of sealing improvements are explained as decreasing of leakage clearance by air jetting. Thus, sealing effects can be improved by amount of jetting air although clearance becomes larger.

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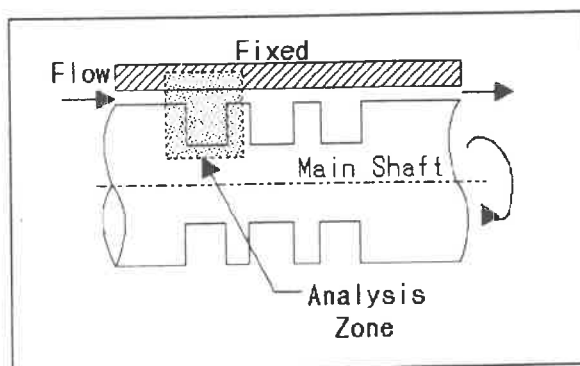


Fig. 1 Non-contact type seal

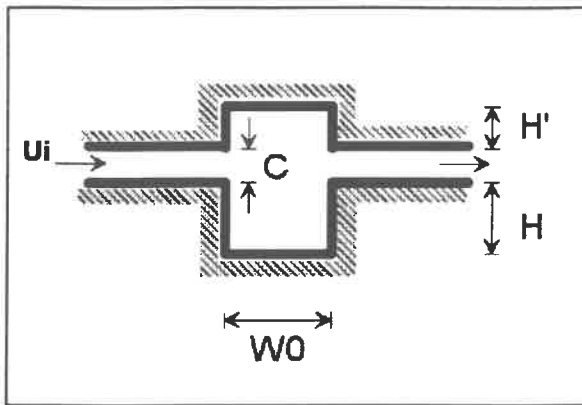


Fig. 2 Design Parameters of Cavity Flow Type

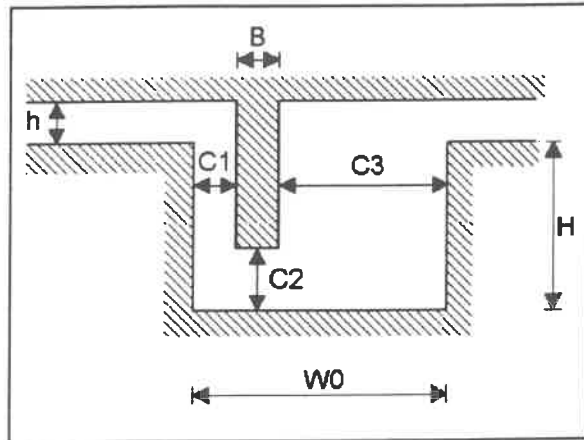


Fig. 5 Design Parameters of Protective Collar Type

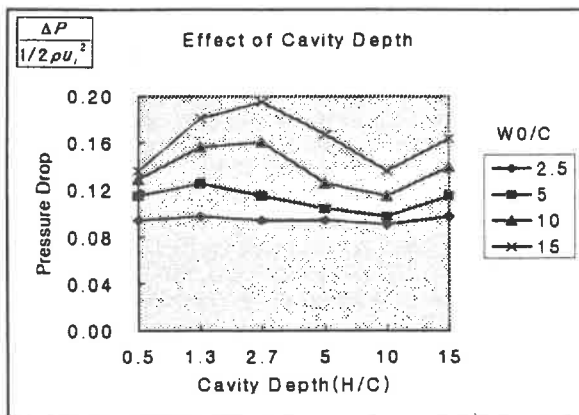


Fig. 3 Effect of Cavity Depth and Width

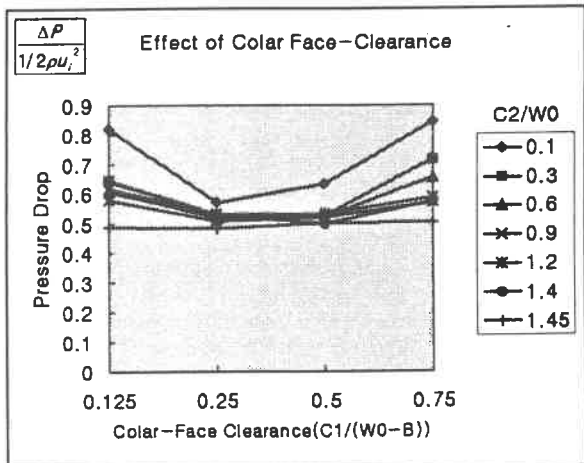


Fig. 6 Effect of Collar Face and End Clearance (C_1 and C_2) on Narrow Passage ($H/W_0=0.1$)

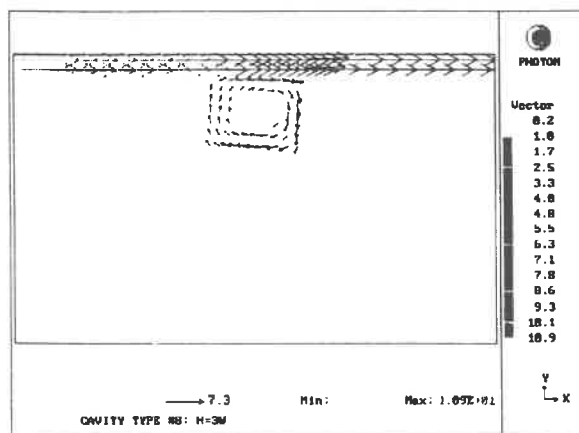


Fig. 4 Dead Circulation in the Cavity

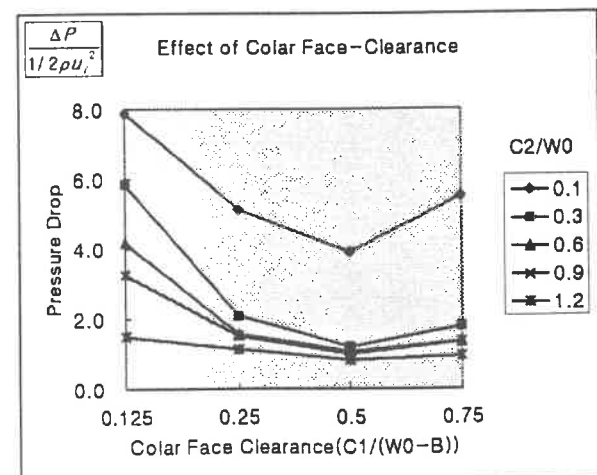


Fig. 7 Effect of Collar Face and End Clearance (C_1 and C_2) on Wide Passage ($H/W_0=0.4$)

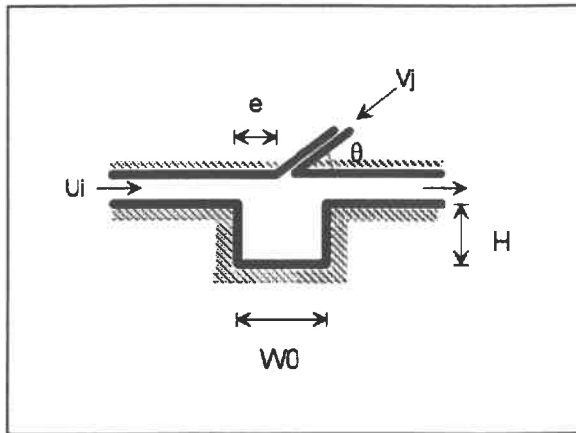


Fig. 8. Design Parameters of Air Jet Type

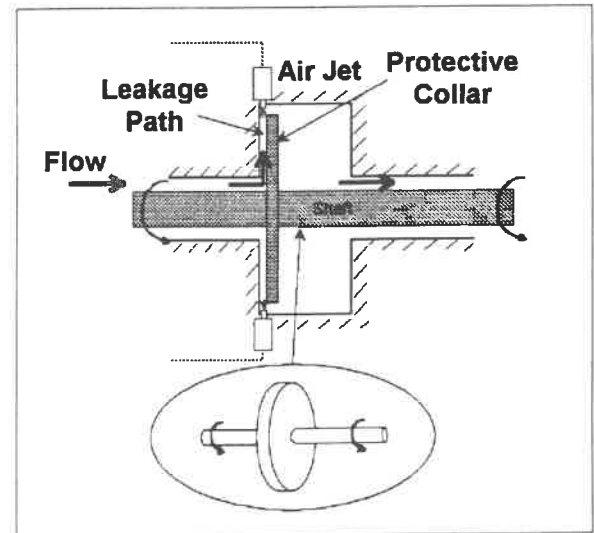


Fig. 11 Adapted Seal Model

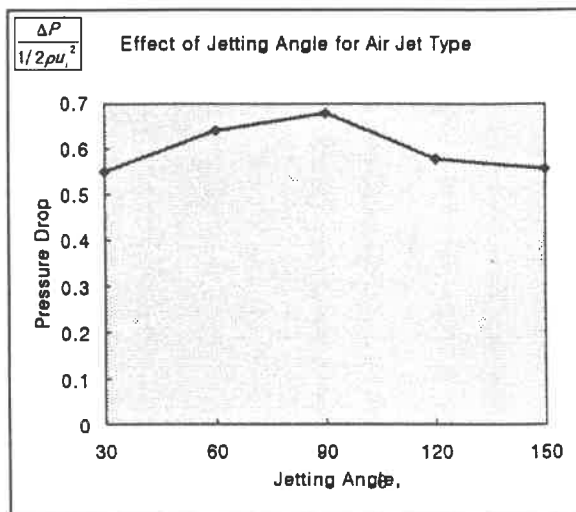


Fig. 9 Effect of Jetting Angle for Air Jet Type

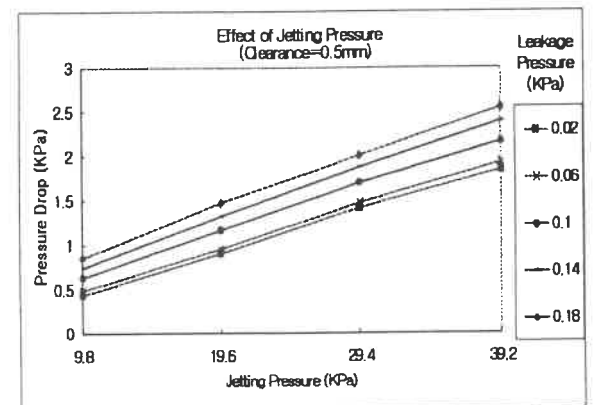


Fig. 12 Effect of Jetting Pressure

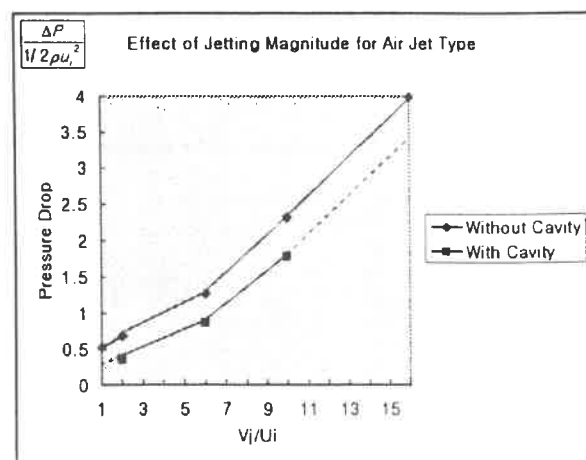


Fig. 10 Effect of Air Jet Magnitude

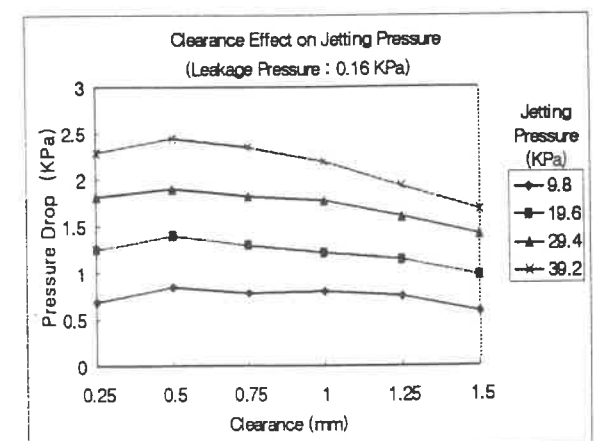


Fig. 13 Effect of Clearance

DERIVATION OF MACHINE TOOL ERROR MODEL FOR 3 AXES VERTICAL MACHINING CENTER USING RIGID BODY KINEMATICS

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ABSTRACT:

Volumetric positional accuracy constitutes a large portion of the total machine tool error during machining. In order to improve machine tool accuracy cost-effectively, machine geometric errors as well as thermally induced errors have to be characterized and predicted for error compensation.

This paper presents the development of kinematic error models accounting for geometric and thermal errors in the Vertical Machining Center (VMC) and a procedure for compensating for geometric and thermal errors in the VMC. The machine investigated is Cincinnati Milacron Sabre 750 3-axes CNC Vertical Machining Center with open architecture controller. Multi-axis machines are typically composed of a sequence of elements (links) connected by joints that provide either rotational or translational motion. Using Rigid Body Kinematics and small angle approximation of the errors, each element of the three axes vertical machining center is modeled using homogeneous coordinate transformation. Given a reference coordinate system and assigning a coordinate frame to machine slides, the orientation and position of the machine slide is expressed in the reference coordinate frame. By synthesizing the machine's parametric errors such as positioning errors, roll, pitch and yaw etc., an expression for the volumetric errors in the multi-axis machine is developed. The developed mathematical model is used to calculate and predict the resultant error vector at the tool workpiece interface for error compensation.

KEY WORDS: CNC machine tool accuracy, thermal and geometrical errors, error modeling, error prediction and compensation.

INTRODUCTION

The accuracy of machined part dimensions depends upon the positional accuracy of the cutting tool relative to the part being machined. Therefore the accuracy of the machine tool used to produce the part is often the limiting factor to obtaining the highest accuracy and part quality. The accuracy of the

machine tool is primarily effected by the geometric errors caused by mechanical-geometrical imperfections, misalignments and wear of the elements of the machine structure, by the non-uniform thermal expansion of the machine structure and static/dynamic load induced errors. The errors can be reduced with the structural improvement of the machine tool through better design and manufacturing practices. However, due to physical limitations, solely production and design techniques can not improve accuracy. Therefore, identification, characterization and compensation of these error sources are necessary to improve machine tool accuracy cost-effectively.

OVERVIEW OF MACHINE TOOL ERRORS

There are three main sources of errors in machine tools that determine machine tool accuracy. These are: 1) Errors due to geometrical inaccuracies, 2) Thermally induced errors and 3) Load induced errors.

1) Errors due to Geometrical Inaccuracies

The mechanical imperfections of the machine tool structure and the misalignment of machine tool elements cause them. They all change gradually due to wear of the components. The effect of the geometrical inaccuracies is to produce errors in the squareness and parallelism between the machine's moving elements. They demonstrate themselves as position and orientation errors of the tool with respect to workpiece.

The 21 geometrical error components of a 3-axes vertical machining center:

	Number of error components
Linear positioning errors (scale error)	3
Straightness errors	6
Angular errors	9
Orthogonality (squareness) errors of machine axes	3
Total	= 21 error components

A Schematic of six error components of the vertical machining center's X-axis carriage system is given in Figure 1. A summary of the symbols used is as follows:

OXYZ: Reference coordinate system	O₁X₁Y₁Z₁: Carriage coordinate system
X: Desired direction of motion	ε_x: Rotational error about X-axis (Roll error)
ε_y: Rotational error about Y-axis (Pitch error)	ε_z: Rotational error about Z-axis (Yaw error)
δ_x: Translational (scale) error along X axis	δ_y: Translational (Horizontal straightness) error along Y axis
δ_z: Translational (Vertical straightness) error along Z axis	

2) Thermally induced errors

They are generated by environmental temperature changes, local sources of heat from drive motors, friction in bearings, gear trains and other transmission devices and heat generated by cutting process.

They cause expansion, contraction and deformation of the machine tool structure and generate positional errors between the cutting tool and workpiece. The machine tool elements particularly effected by self-generated thermal distortion are spindles and leadscrews.

3) Load induced errors

There are three different types of forces present during machining process: 1) Workpiece weight
2) Forces resulting from cutting process 3) Gravity forces resulting from the displacement of masses of the machine components. They all cause elastic strain on the machine tool structure.

HOMOGENEOUS COORDINATE SYSTEM TRANSFORMATION OF MACHINE TOOL CARRIAGE-GUIDE SYSTEM

Multi-axis machines are typically composed of a sequence of elements (links) connected by joints that provide either rotational or translational motion. Using Rigid Body Kinematics, each element and joint can be modeled as a homogeneous coordinate transformation. A homogeneous transformation matrix in 3 dimensional space is a 4 x 4 matrix. It can be used to represent one coordinate system with respect to another or reference coordinate system. A general homogeneous transformation matrix with null perspective transformation is in following form:

$$T = \begin{bmatrix} R_{3 \times 3} & P_{3 \times 1} \\ f_{1 \times 3} & 1 \times 1 \end{bmatrix} = \begin{bmatrix} \text{Rotation Matrix} & \text{Position Vector} \\ \text{Perspective Transformation} & \text{Scaling} \end{bmatrix}$$

$$T = \begin{bmatrix} n_x & s_x & a_x & P_x \\ n_y & s_y & a_y & P_y \\ n_z & s_z & a_z & P_z \\ 0 & 0 & 0 & 1 \end{bmatrix} = \begin{bmatrix} n & s & a & p \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad \text{where:}$$

n, s, a: Orientation of a coordinate system with respect to another coordinate system.

p: Position vector of the origin of a coordinate system with respect to another coordinate system.

If the coordinate frame is embedded in an object, using homogeneous transformation matrix it is possible to describe relative position and orientation of this object with respect to another object or coordinate system in space. They can be multiplied in series to describe one object with respect to several different coordinate frames. A homogeneous transformation matrix describes the pure translation of an ideal carriage in the following form:

$$T_I = \begin{bmatrix} 1 & 0 & 0 & X \\ 1 & 0 & 0 & Y \\ 1 & 0 & 0 & Z \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad \text{where}$$

X: Variable position of carriage coordinate system origin with respect to the reference coordinate system.

Y, Z: Constant offsets of the origin of carriage coordinate system origin with respect to the reference coordinate system.

In reality, any single degree of freedom carriage has error motions in six degrees of freedom. The total error motion of the carriage is a combination of rotational and translational errors. The homogeneous transformation matrix of rotational errors is given below with the assumption of small angular errors:

$$T_{\text{rot}} = \begin{bmatrix} 1 & -\epsilon_z & \epsilon_y & 0 \\ \epsilon_z & 1 & -\epsilon_x & 0 \\ \epsilon_y & \epsilon_x & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

The total translational error matrix is given as:

$$T_{\text{trans}} = \begin{bmatrix} 1 & 0 & 0 & \delta_x \\ 1 & 0 & 0 & \delta_y \\ 1 & 0 & 0 & \delta_z \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

The total error matrix is (neglecting second order terms) is:

$$E = \begin{bmatrix} 1 & -\epsilon_z & \epsilon_y & \delta_x \\ \epsilon_z & 1 & -\epsilon_x & \delta_y \\ \epsilon_y & \epsilon_x & 1 & \delta_z \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

The actual position and orientation of the carriage in reference coordinate system can be calculated as following:

$${}^R T_a = {}^R T_i E = \begin{bmatrix} 1 & -\epsilon_z & \epsilon_y & \delta_x + X \\ \epsilon_z & 1 & -\epsilon_x & \delta_y + Y \\ \epsilon_y & \epsilon_x & 1 & \delta_z + Z \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

DERIVATION OF ERROR MODELS FOR SABRE 750 3 AXES VMC

As an example, the actual position and orientation of the Y-axis carriage (Figure 2) in reference coordinate system is given as follows:

$${}^R T_1 = \begin{bmatrix} 1 & -\epsilon_{z1}(y) & \epsilon_{y1}(y) & \delta_{x1}(y) \\ \epsilon_{z1}(y) & 1 & -\epsilon_{x1}(y) & \delta_{y1}(y) + b_1(y) \\ \epsilon_{y1}(y) & \epsilon_{x1}(y) & 1 & \delta_{z1}(y) + c_1 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad \text{where}$$

$\epsilon_{y1}(y)$: Roll error of Y-axis. $\epsilon_{x1}(y)$: Pitch error of Y-axis.

$\epsilon_{z1}(y)$: Yaw error of Y-axis.

c_1 : Constant offset in Z direction between O and O₁.

$b_1(y)$: Offset in Y direction between O and O₁.

$\delta_{y1}(y)$: Linear displacement error of Y axis.

$$\delta_{x1}(y) = \delta'_{x1}(y) + \alpha_{xy} \delta y$$

$\delta'_{x1}(y)$: X straightness of Y-axis as it moves in Y direction.

α_{xy} : Orthogonality error between X and Y axes.

Δy : Incremental Y motion that amplifies α_{xy} to yield an Abbe error in X direction.

$$\delta_{z1}(y) = \delta'_{z1}(y) + \delta_{zy} \Delta y$$

$\delta'_{z1}(y)$: Z straightness of Y-axis as it moves in Y direction.

δ_{zy} : Orthogonality error between Y and Z axes.

Δy : Incremental Y motion that amplifies α_{zy} to yield an Abbe error in Z direction.

$${}^R T_{\text{Work}} = {}^R T_1 {}^1 T_2$$

$${}^R T_{\text{Tool}} = {}^R T_3$$

$T_{\text{Work}} = T_{\text{Tool}} E$. Therefore, errors can be calculated using following matrix equation:

$$E = {}^R T_{\text{Tool}}^{-1} {}^R T_{\text{Work}}$$

$$\begin{bmatrix} P_x \\ P_y \\ P_z \end{bmatrix}_{\text{Correction}} = \begin{bmatrix} P_x \\ P_y \\ P_z \end{bmatrix}_{\text{Work}} - \begin{bmatrix} P_x \\ P_y \\ P_z \end{bmatrix}_{\text{Tool}}$$

$$P_{\text{Correction}} = P_{\text{Work}} - P_{\text{Tool}} = P({}^R T_1 {}^1 T_2) - P({}^R T_3)$$

$$P_{X\text{Correction}} = \delta_{x2}(x) + a_2(x) - \epsilon_{z1}(y) [\delta_{y2}(x) + b_2(y)] + \epsilon_{y1}(y) [\delta_{z2}(x) + c_2] + \delta_{x1}(y) - [\delta_{x3}(z) + a_3]$$

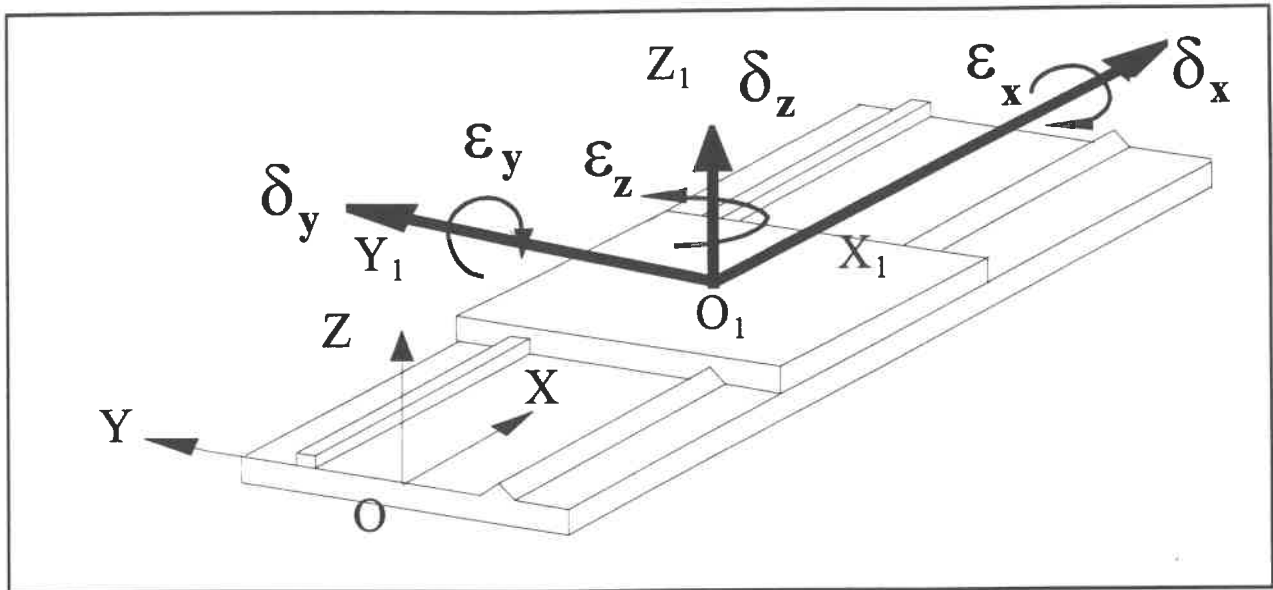


Figure 1. Schematic of six degrees of freedom error motion of a machine tool carriage system

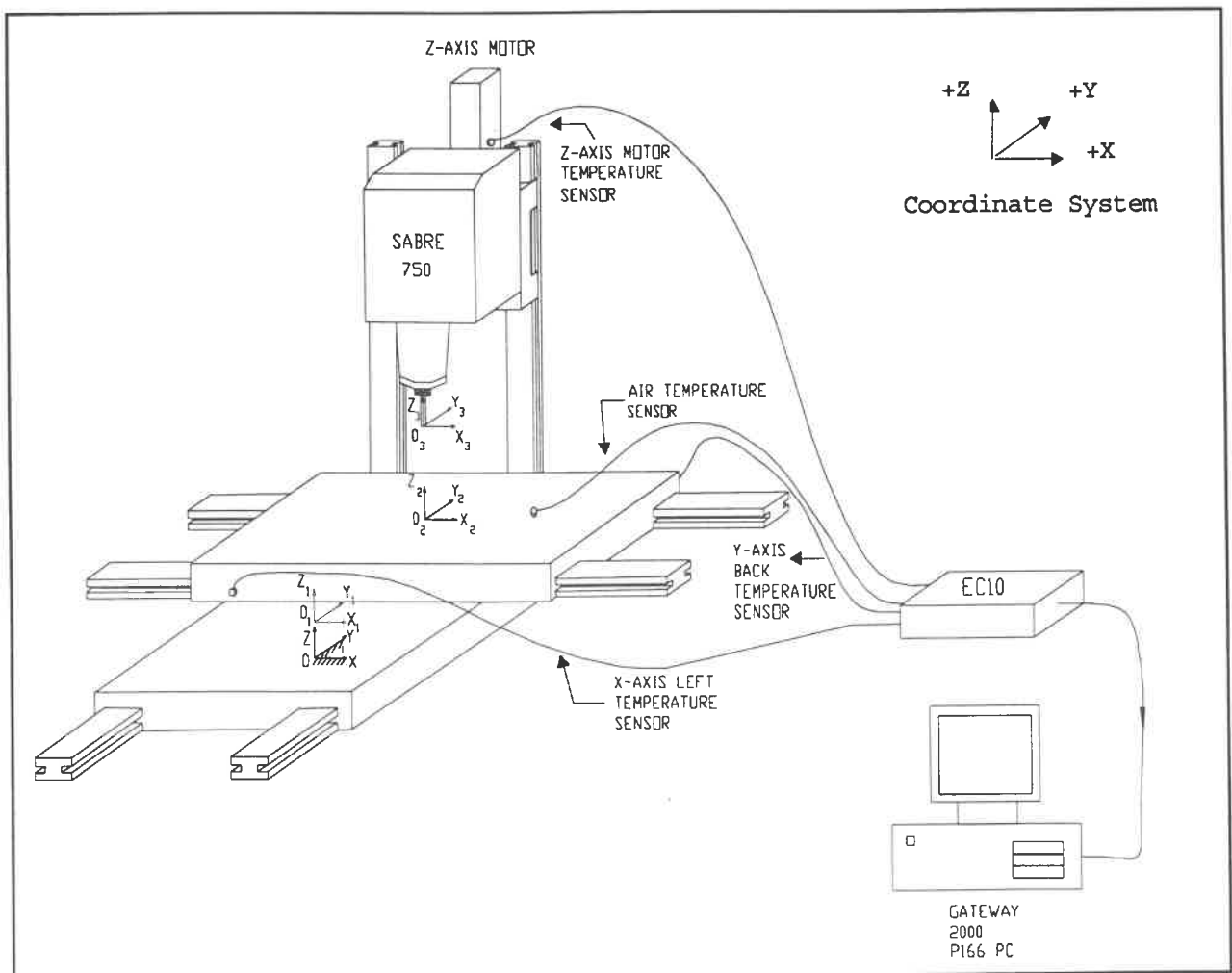


Figure 2. Schematic of the Sabre 750 VMC Layout.

$$\begin{aligned}
P_{Y\text{Correction}} &= \delta_{y2}(x) + b_2(y) + \epsilon_{z1}(y) [\delta_{x2}(x) + a_2(x)] - \epsilon_{x1}(y) [\delta_{z2}(x) + c_2] + \\
&\quad \delta_{y1}(y) + b_1(y) - [\delta_{y3}(z) + b_3] \\
P_{Z\text{Correction}} &= \delta_{z2}(x) + c_2 - \epsilon_{y1}(y) [\delta_{x2}(x) + a_2(x)] + \epsilon_{x1}(y) [\delta_{y2}(x) + b_2(y)] + \\
&\quad \delta_{z1}(y) + c_1 - [\delta_{z3}(z) + c_3(z)]
\end{aligned}$$

CONCLUSION

A procedure for the modeling and compensation of Vertical Machining Center's volumetric errors has been demonstrated. The actual implementation of the above model will be presented in detail in subsequent paper.

ACKNOWLEDGEMENT

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Modeling Nonlinear Behavior in a Piezoelectric Actuator

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Abstract

Outline

Piezoelectric actuators have become a standard option in positioning applications where the displacements must be small and highly accurate. In particular, nanometric cutting demands exceptionally fine and repeatable motions under significant environmental perturbations. In addition to perturbations, several nonlinearities are present in piezoelectric actuators. The well-known phenomena of hysteresis and creep, along with nonlinear voltage dependence affect the dynamic response, making operation under feedback control a primary need [1]. Consequently, precise mathematical models are necessary. Considerable research has been reported in the literature that describes the linear portion of the behavior of piezoelectric actuators, especially for the stacked-type construction. Wang and co-workers [2] analyzed the stack using a structural approach, with applications to vibration isolation and sensing; Main and Garcia [3] provided an interesting evaluation of several modeling approaches, including models that are able to describe hysteresis. Several works have been reported that specifically address the nonlinearities. For example, Goldfarb and Celanovic [4] utilized a Maxwell slip model to describe hysteresis. The creep phenomenon that is observed in piezoelectric materials has been treated in much fewer publications. Moreover, these works mostly deal with descriptions of creep that are not intended to serve as models for control applications, but are merely empirical time-based correlations [5]. In this paper, a mathematical model is presented that describes both creep and hysteresis, as well as nonlinear voltage dependence. The model is expressed in differential form, making it suitable for later controller synthesis. Experimental validation of the model is also presented. The experiments were performed in a prototype nanometric cutting instrument, which employs a PZT tube as an actuator. The PZT tube in this instrument can be positioned in three dimensions with a theoretical sub-nanometer resolution, but the effects mentioned above prevent achieving accuracy in the same order of magnitude. A suitable mathematical model, along with a feedback control scheme may expand the capabilities of the instrument to be used as a nanometric machine tool. In this work, the uniaxial extension case is modeled using a non-formal analogy between nonlinear viscoelasticity and the observed behavior. In this analogy, the voltage applied to the actuator is treated as axial stress, and the piezoelectric effect is modeled as a simple proportionality relation between voltage and axial stress. This way, the nonlinear piezoelectric effects, which are difficult to model, are transferred to the more manageable nonlinear viscoelastic analysis. Even though the physical grounds for such an analogy are not defined, the model can be regarded as an empirical *dynamic* representation that is useful for further developments. Future work in this direction includes an extension of the model to the three-dimensional case and the inclusion of an arbitrary point load representing the cutting force. The development of control strategies for both the uniaxial case and the general three-dimensional case is also a future goal.

Three Nonlinear Effects

The block diagram in Fig. 1 shows the setup employed for all experimental work. The z-axis open-loop positioning system consists of an input source, which can be a function generator or a personal

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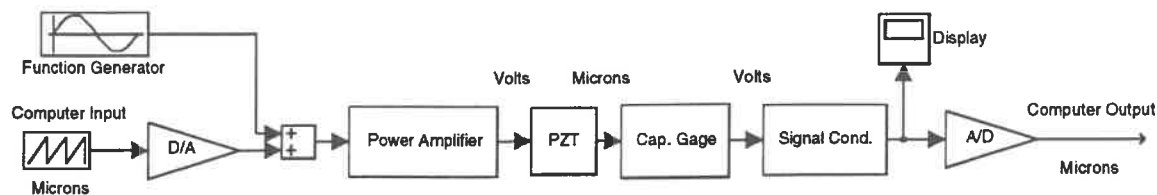


Figure 1

computer, a high voltage amplifier, the piezoelectric tube actuator, and a capacitive position sensor. The signal of the position sensor is read by a personal computer or a digital oscilloscope with data storage capability. Both analog/digital converters are 16 bit devices. The capacitance gauge and signal conditioner have a joint sensitivity of $3.96 \text{ V}/\mu\text{m}$, resulting in an A/D resolution of 0.077 nm/bit . The PZT tube and power amplifier have an average ² joint sensitivity of $0.15 \mu\text{m/V}$, resulting in a D/A resolution of 0.046 nm/bit . These resolutions are far from being fully effective in practice due to the nonlinear effects under study. The well-known effects of nonlinear gain and steady creep under constant input voltage, in addition to hysteresis under cyclic excitation, are experimentally confirmed and then used for identification of the model parameters. Figure 2 (left) shows a set of normalized³ responses to a voltage step input for various voltage levels. The non-uniqueness of the normalized response suggests the presence of a nonlinear input gain. The same experiment, but in a much larger time scale, is shown in Fig. 2 (right), where a constant rate of creep is observed. It should be noted that the curves are normalized, indicating that creep is nonlinear with respect to input voltage. A series of triangular voltage inputs are applied to the system, resulting in the loop plots depicted in Fig. 3. It is important to state that the plots reflect the superposition of two effects, namely, *true* rate-independent hysteresis and phase lag due to system dynamics, which is obviously frequency-dependent. In most cases, both effects appear coupled [6], and the choice of decoupling or not in a model identification procedure depends on the particular case under study. Throughout this work, the term "hysteresis" is used to denote the coupled phenomenon rather than either of the individual components.

The Proposed Model

A model is derived based on an analogy that relates the observed behavior with the one corresponding to nonlinear viscoelastic materials. Such materials exhibit an input/output behavior that is remarkably similar to the nonlinear piezoelectric case, in a qualitative sense. In the analogy, stress and strain in viscoelastic materials are replaced by voltage and tip displacement in the piezoelectric case. The model equations below, in state-space form, are constructed based on a generalization of a Burgers rheological model, which consists of an arrangement of springs and nonlinear dashpots.

$$\begin{cases} \dot{u}_1 = -\lambda_1 u_1 + \lambda_1 r_1 f_t(V(t)) \\ \dot{u}_2 = -\lambda_2 u_2 + \lambda_2 r_2 f_t(V(t)) \\ \vdots \\ \dot{u}_n = -\lambda_n u_n + \lambda_n r_n f_t(V(t)) \\ \dot{u}_s = f_s(\sigma(t)) \end{cases} \quad (1)$$

The output equation is

$$u = u_1 + u_2 + \cdots + u_s + f_e(V(t)) \quad (2)$$

²The PZT tube has a nonlinear static sensitivity.

³Divided by the value of the input

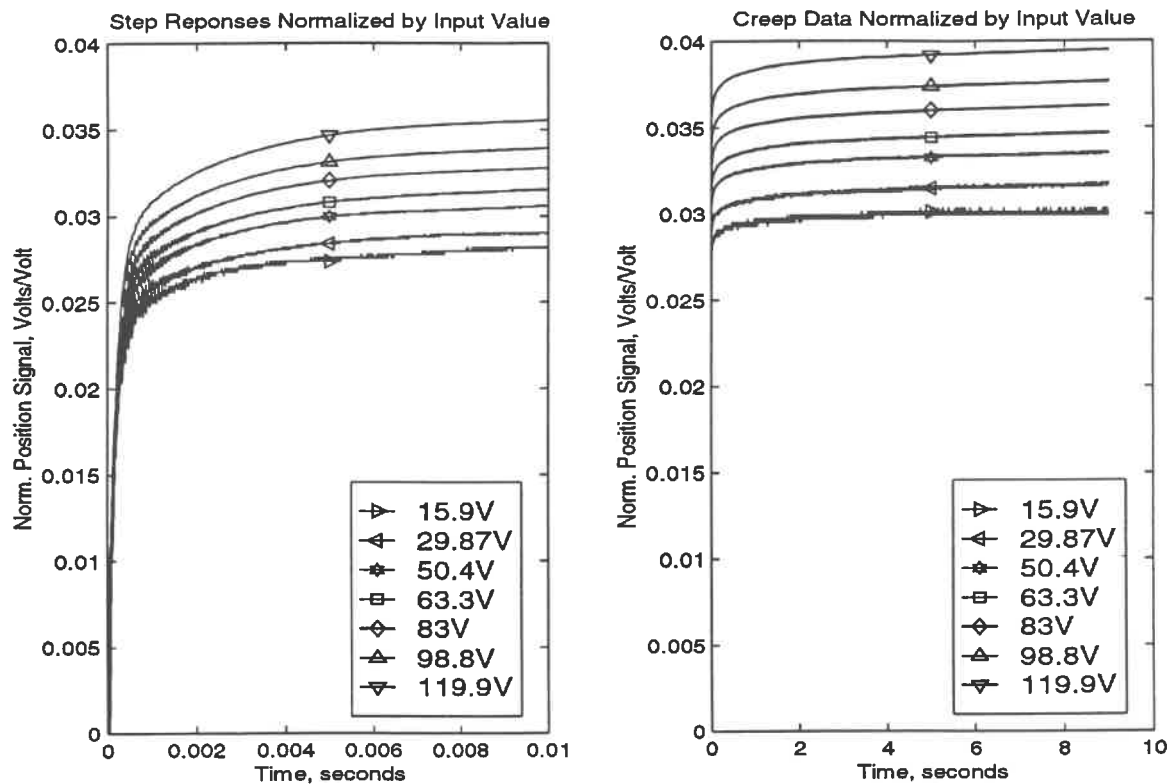


Figure 2

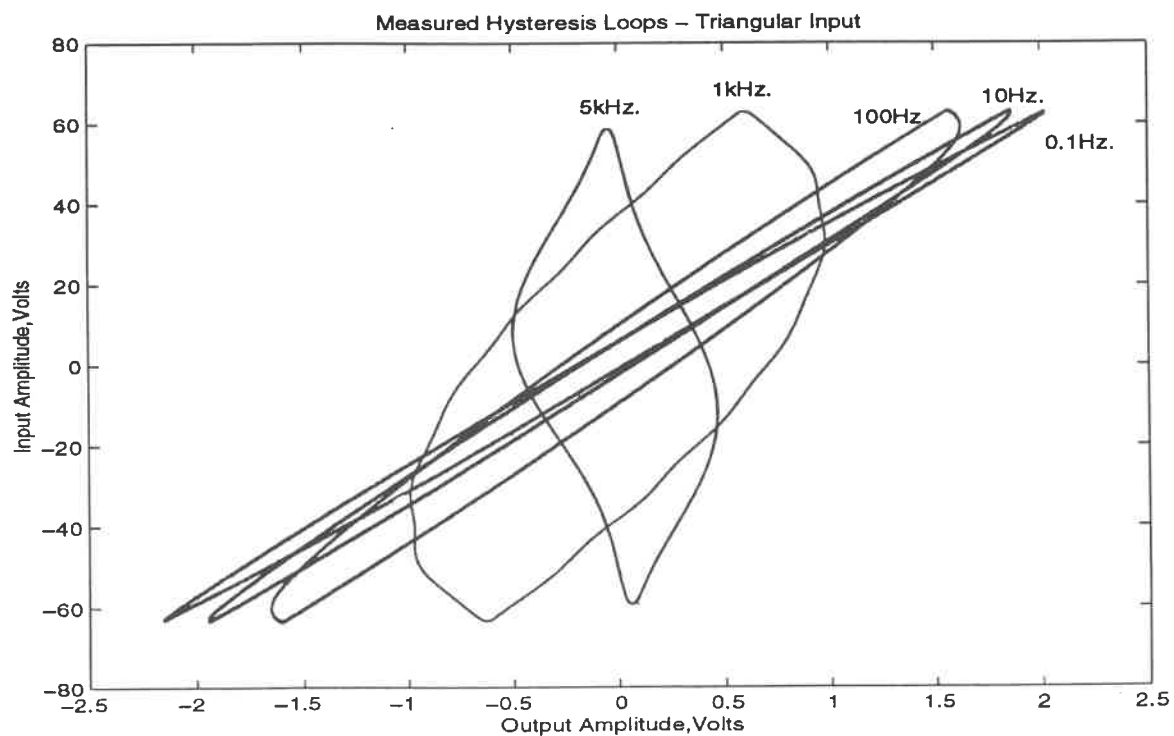


Figure 3

where the first n u_i states are the contributions to tip displacement from each parallel spring-dashpot arrangement in a Burgers model, and u_s is the state that describes steady creep. $V(t)$ and $u(t)$ denote input and output respectively. The functions f_t and f_s can be arbitrarily chosen to match the nonlinear dependence on input voltage for transient displacement and steady creep, respectively. Power and hyperbolic sine laws are common choices in nonlinear viscoelasticity. In the present work, a power law of voltage is chosen for both functions, and have good results. The f_e function is originally intended to represent instantaneous elastic response in viscoelastic materials and corresponds to the isolated spring in a Burgers model. In the present case, it is understood that there is no instantaneous response of any sort, so its use is precluded. Parameters λ_i correspond to the inverse of retardation time in the viscoelastic case, and the constants r_i weigh the contribution of each Burgers element to the overall transient displacement. The weighting factors must sum to unity. These parameters, along with the ones that may be included in the functions f_t and f_s are to be recovered from a set of experiments by any suitable method.

Parameter Identification Procedure

The following laws are chosen for the input functions:

$$f_t(V) = \left(\frac{V}{\mu}\right)^p$$

$$f_s(V) = \left(\frac{V}{\eta}\right)^m$$

where p and n are, in the most general case, non integer positive quantities, often satisfying $1 < m < p$. When the input is allowed to take on negative values, slight modifications must be introduced to keep the functions from having imaginary values. The parameters introduced by these functions, along with the ones defined earlier, are identified from a set of experiments in the time and frequency domain. More explicitly, η, μ, p and m are directly obtained from the set of step input responses depicted in Figs. 2, while the remaining parameters are recovered using both time domain and frequency domain data.

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AN EXPERIMENTAL INVESTIGATION INTO THERMAL SPINDLE GROWTH UNDER LOAD

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Keywords: CNC vertical machining center, Loading device, Spindle error analyzer, and Thermal spindle stability

Introduction

Current measurement methods specified by ANSI/ASME B5.54 standard¹ for spindle thermal growth establish growth characteristics with no load applied to the spindle bearings. It is possible that the application of load to the spindle bearings will result in either different heating, different response to the heating or both.

This paper describes the design and construction of a loading device to simulate thrust loads on the spindle bearing system of a Monarch computer numeric control (CNC) vertical machining center. It also discusses results of loaded and unloaded thermal spindle growth at various loads and Rpm's. All tests were conducted according to the methods specified for thermal spindle stability tests in ANSI/ASME B5.54: A commercially available spindle error analyzer³ (SEA) system with capacitance gauging techniques was used to measure axial, radial, and tilt thermal drift of the spindle.

Loading device concept and design

A loading device, an external fixture that applies axial loads on the spindle in much the same way as in a typical cutting process, was designed and built in order to apply and monitor the effects of various loads on the spindle. The advantage of such an external device is that it can be easily mounted on most CNC vertical or horizontal spindle machining centers with little or no modification to the existing machine tool design. It can also be installed and removed quickly for comparison study of a spindle under loaded and unloaded conditions. The loading device was designed and built to include the following features:

- Simulate axial loads
- Allow load variations
- Adapt to the design of existing machines
- Isolate the device from the metrology loop of the machine

In order to meet the above requirements, high-energy rare earth permanent magnets were used. The development of these magnets concentrates a large amount of magnetic energy into a small package, making very efficient use of available space. These magnets also offer much higher performance characteristics in terms of energy product, thermal operating range, and stability.

Experimental setup

The experimental setup of the loading device is shown in Figure 1 (SEA system not shown). To improve repeatability of measurements, the magnet mount plate and the target plate were not

removed during the unloaded thermal spindle stability tests. A brief description of the loading device components and their functions is given below.

Neodymium magnets and Magnet mount plate

The loading device utilizes a pair of rare-earth Neodymium magnets, mounted an equal distance from the centerline of the spindle. The primary function of the magnet mount plate is to provide a stable surface for the magnets and to keep eddy current losses to a minimum. The plate is attached to the spindle bracket of the machine.

To avoid damage to the plate or machine components during power failure or during referencing of machine, a 10-degree taper is provided on two sides of the plate. Another advantage of this taper is that it avoids any interference with the tool changer bracket mounted closely to the spindle. Magnet mount plate is

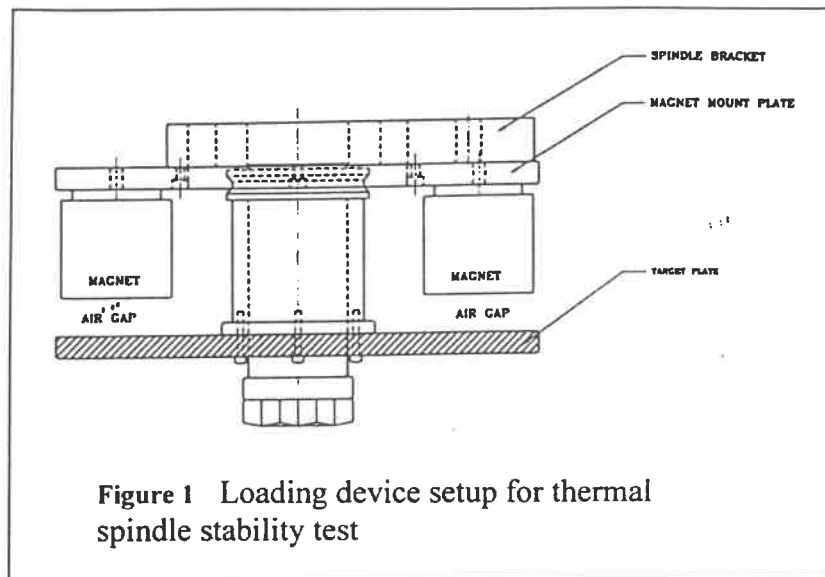


Figure 1 Loading device setup for thermal spindle stability test

made of 0.50-inch thick aluminum. This not only avoids fringing but also minimizes deflection of the plate when the magnets are mounted and applying load to the spindle.

Extended Shank Collet Chuck and Precision Spacers

An extended shank collet chuck is required to provide several variations of gap when a tool holder is mounted into the spindle and the magnet mount plate is mounted to the spindle bracket. To facilitate easy loading and unloading of the spindle, a steel sleeve with tapped holes for 10-32 SHCS is heat-shrunk on the straight shank of the collet chuck. When the collet chuck is mounted into the spindle, tapped holes at the bottom of the sleeve allow quick mounting of the target plate. Precision ring-spacers of various thicknesses are used to vary the gap between magnet pole faces and the target plate. To minimize heat transfer from the target plate to the collet chuck, the spacers are relieved at their inner edge.

Target Plate and Aluminum Flats

The target plate as shown in Figure 2 completes the magnetic flux loop, which applies an axial load to the spindle. Magnetic flux from one pole of the magnet goes through the plate and then returns to the opposite pole of the magnet. The target plate is made of 1010 steel, chosen because its permeability is comparable to a magnetically soft material, yet is cost-effective and widely available. Initially, tests with a *solid* target plate showed that there was a significant rise in target plate temperature compared to that of the lower bearing. One effective method of reducing temperature rise in the plate and elsewhere in the loading device is to cut down on eddy current paths.

The target plate was modified using electrodischarge machining (EDM) to cut radial slots into the plate to interrupt the current paths. In addition to the slots, numerous holes were drilled in the plate to increase convection heat transfer and minimize conduction paths from the plate to the collet chuck. Material at the inner periphery was also removed to insulate the collet chuck from eddy current heating.

Initially, two aluminum rectangular flats as shown in Figure 2 were used to monitor the temperature rise of the target plate due to hysteresis and eddy currents power loss. In the eddy current test setup magnets were mounted on the flats and machine's Z-axis was moved to set a test gap between magnet pole faces and the target plate.

Estimation and measurement of axial forces

Different cutting processes require varying machining forces and power. The magnitude of forces generally found in actual cutting operations is on the order of a few hundred Newton². Based on sample calculations and a limiting value of magnetic forces at a safe allowable gap, a maximum limit of about 140-lbs. (623 N) was set. An S-type load cell with a bridge resistance of 350 ohms was used to measure forces applied by magnets.

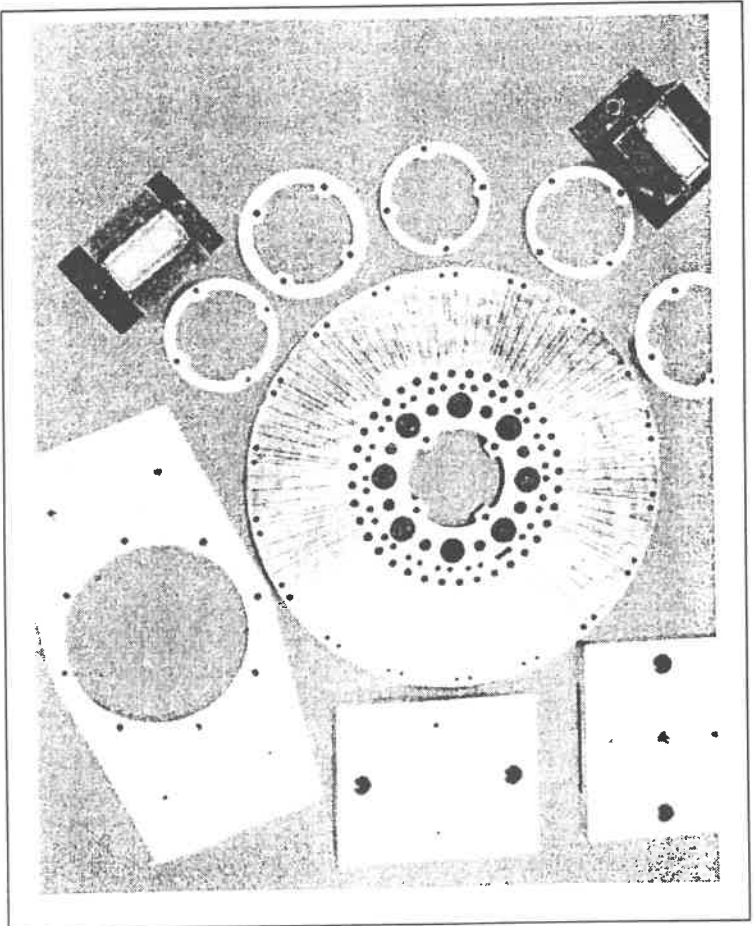


Figure 2 Loading device components

Compensating for thermal drift due to hysteresis and eddy current power loss

When the spindle is running at operating speed with the loading device installed, there are two types of power losses evident in the target plate; hysteresis and eddy current. This dissipation of power generates external heat raising the temperature in the experimental set up for monitoring the loaded thermal spindle growth. Since this heat source is external, it must be separated from the spindle thermal drift under loaded conditions. To isolate the temperature rise due to eddy current and hysteresis power loss, thermistors placed at various locations on the loading device monitor the temperature.

The spindle was stopped periodically to record the temperature. By taking the average temperature rise of the tool holder and SEA master target, along with corresponding thermal coefficients of expansion, a time versus drift relationship was obtained for a given RPM and load. This drift was then subtracted from the loaded thermal spindle growth data to obtain spindle drift under loading conditions. Maximum growth of 5 microns was obtained at 1000 RPM and under the application of 140 lbs. of axial load.

Correlating unloaded and loaded thermal spindle growth

In preface to a discussion of loaded and unloaded thermal spindle growth, here are some pertinent specifications about the machine tool spindle

The Monarch vertical machining center has tapered roller bearings supporting the spindle. At high speeds, the spindle bearing preload is automatically reduced by a hydraulic integrated circuit (HIC). From 0-525 RPM, pressure to spindle bearing is 400 psi, and, from 525-1125 RPM, pressure to the spindle bearing is 100 psi.

Loaded and unloaded thermal spindle growth at 255 RPM

Initially (about one hour), spindle drift remained constant regardless of the load. At no load, there was a sharp rise in the drift as the bearings heated up from ambient. After about five hours, drift reached equilibrium. Similar behavior of the spindle drift could be observed at two different loads. Although spindle

thermal drift-rate was same under two significantly different loads, its magnitude was approximately four microns higher when compared with drift under no load.

The time constant was approximately the same with no load to the spindle or with application of 50 lbs. of

load. However, the time constant was slightly higher when load was increased to 140 lbs. More heat generated in the bearings by the applied axial load may have caused this longer time constant and a larger drift-rate.

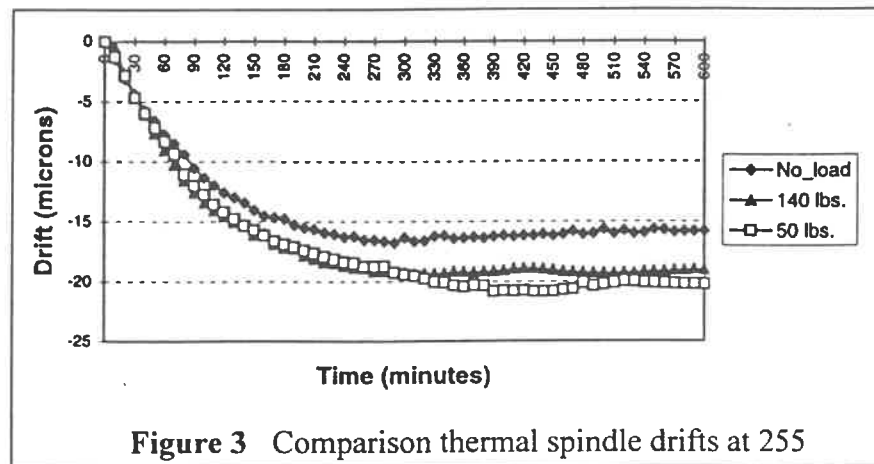


Figure 3 Comparison thermal spindle drifts at 255

Loaded and unloaded thermal spindle growth at 500 RPM

For the first hour, there was no significant difference in the magnitude of drift with load or no

load. After about two hours, the drift varied consistently with time till equilibrium was reached.

This indicated that during machine start-up, thermal spindle growth behaved the same regardless of the load but changed as much as 6-8 microns depending upon the amount of load applied to the spindle bearings. The time constant was the same;

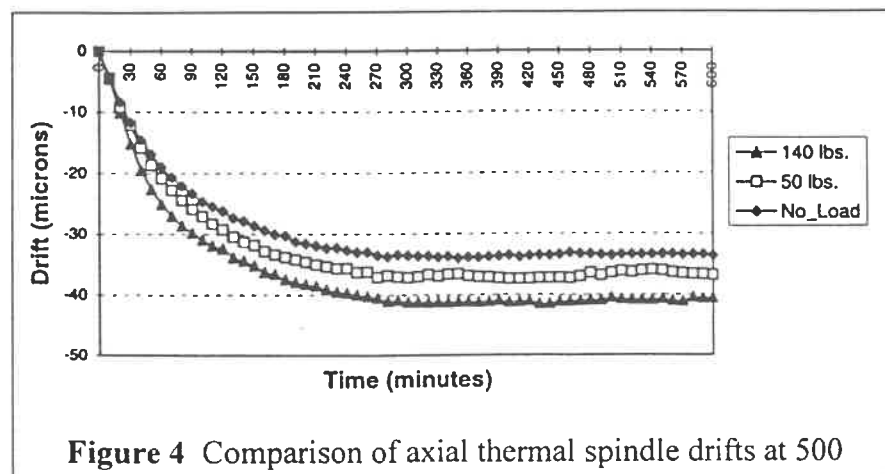


Figure 4 Comparison of axial thermal spindle drifts at 500

approximately four and one half hours of either no load or 50 lbs. of load.

However, the time constant was slightly higher when load was increased to 140 lbs. A small difference in time constants at 500 RPM may have been caused by more heat generation in the bearings under the application of different loads. Also, as the applied spindle load was increased, thermal drift rate increased proportionally. An upward shift of 1-2 microns towards the end of the test was caused by the structural drift of the machine, in a direction opposite to that of the spindle drift. This also indicates that the machine structure has a longer time constant than the spindle system of the machine.

Loaded and unloaded thermal spindle growth at 1000 RPM

As shown in Figure 5, spindle drift behavior is similar under no load or two significantly different loads. During the initial warm-up period, spindle growth and thermal drift-rate under

load and no load remained same. After three hours, however, drift varied about 2-3 microns. This difference in drift under load and no load conditions was about the same as the variance in the measurements. Therefore, it could not be considered significantly large. The time constant at this RPM was approximately four and one half hours under the

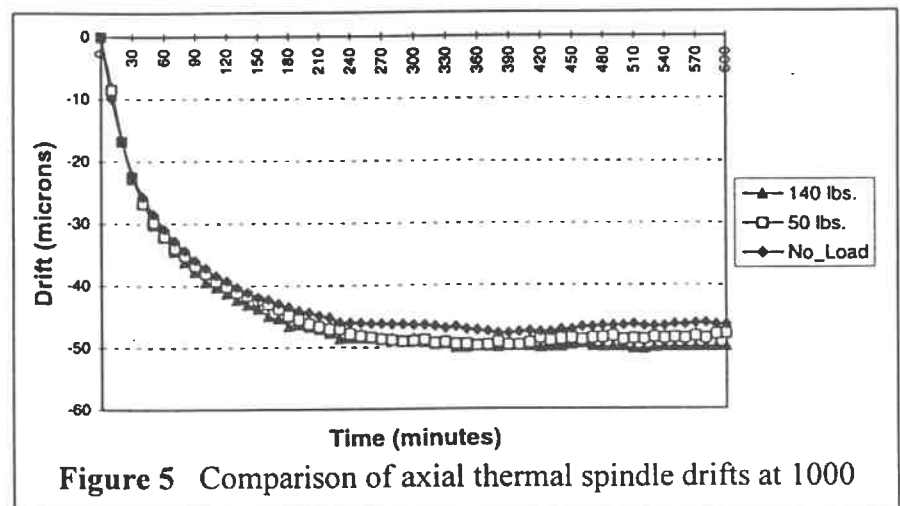


Figure 5 Comparison of axial thermal spindle drifts at 1000

application of 50 lbs. of load. A slight increase in the time constant was obtained under the application of 140 lbs. of load. At this RPM external axial loads on the spindle bearing do not influence thermal drift rate. Some fluctuations in the drifts are possible due to small changes in ambient temperature and structural drift of the machine under heat.

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A CONTRIBUTION TO SYNTHETIZATION ERRORS MODELS OF MACHINE TOOLS

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INTRODUCTION

The CNC machine tools undergo the effects of several internal and external factors that affect the workpiece dimensional accuracy since they modify the commanded tool and workpiece positions. This error position is usually called as machine total error. Analyzing the movements of each machine element and its respective errors, it is possible to model mathematically the machine structure for obtaining the total error.

The knowledge of the total error permits the development of an error compensation system that provides more accurate working conditions. A compensation system needs information about the value and behavior of all the machine parametric errors (translation and rotation errors), that can be supplied by an adequate calibration procedure. However, the parametric errors change during the machine operation. When the machine is working, it experiments several temperature states that provoke some structure deformations that modify the parametric error behavior [1]. Therefore, the parametric error calibration should be done positioning the machine carriages in several points within the machine working domain and until the temperature and the error value stabilize [5,7]. This calibration procedure creates a large deal of data which must be statistically analyzed to obtain an expression for the total error in any position and machine thermal state.

The actual work presents, in a simple form, a mathematical model of the machine tools parametric errors and of its variations due thermal effects, that minimizes the data volume obtained during the calibration and predicts the machine total error. This methodology was applied to the prediction of the thermal errors in a CNC cylindrical grinding machine. The simulated results were evaluated and compared with the errors measured in workpieces.

MATHEMATICAL MODEL

The error expressions for the evaluated machine were obtained using Homogeneous Transformation Techniques to describe the relative position between the tool and the workpiece [2,6]. The model combines adequately the individual errors in each preferential direction, X and Z, of the grinding machine. The total error can be called too, in this case, as “planar error”. The modeled machine and the several coordinate systems associated to its elements are shown in the figure 1. The tool path, with respect to the reference system 0, is given by the systems 1,2,3 and 4 and it can be written as the matrix T_{TOOL} . On the other side, the piece path with respect to system 0, is described by the adequate combination of the systems 5, 6 and 7 and is written as the matrix T_{PIECE} :

$$T_{TOOL} = {}^0T_1 \cdot {}^1T_2 \cdot {}^2T_3 \cdot {}^3T_4 \quad T_{PIECE} = {}^0T_5 \cdot {}^5T_6 \cdot {}^6T_7 \quad (1)$$

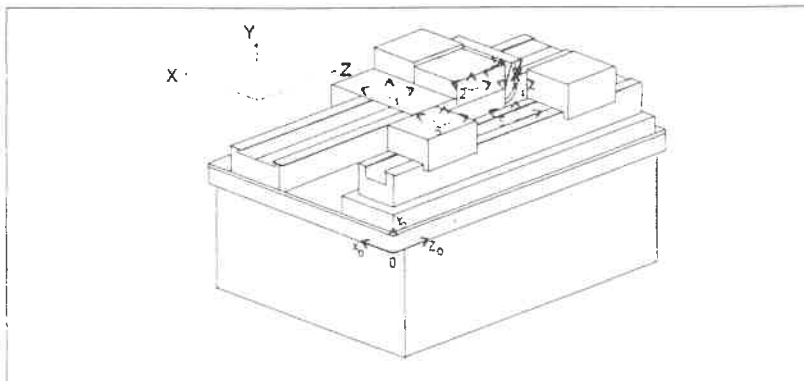


Figure 1: Coordinate Systems in the CNC cylindrical grinding machine

After the necessary matrix operations and the elimination of the terms of second or higher order [4], the mathematical error expressions, (2) and (3), for the components, E_x and E_z , of the planar error are obtained.

$$E_x = \delta_x(x) + \delta_x(z) - \varepsilon_z(z) \cdot Y_{12} + \varepsilon_y(z) \cdot (Z_{12} + Z_{23}) + \varepsilon_y(x) \cdot Z_{23} \quad (2)$$

$$E_z = \delta_z(z) + \delta_z(x) - \varepsilon_y(z) \cdot (x - X_{12} + X_{23} - X_{34}) - \varepsilon_y(x) \cdot (X_{23} - X_{34}) + \varepsilon_x(z) \cdot Y_{12} \quad (3)$$

where

$\delta_i(j)$ are translation errors in the direction i due to the displacement in the direction j ; $\varepsilon_i(j)$ are rotation errors about the axis i due to the displacement in the direction j ; X_{kl} , Y_{kl} , Z_{kl} are offsets between the coordinate systems k and l in the directions X , Y and Z , respectively and x is the coordinate in the X direction.

The component E_x of the planar error is of major interest since the direction X is responsible for the definition of the piece diameter (main piece dimension). In the Z direction, the workpiece has usually larger tolerances. The parametric error calibrations were made considering the different temperature states of the machine. With this propose an automatic data acquisition systems was developed. The systems allows to control and to command sequential lectures of the actual temperature and the parametric error. The temperature was monitored through temperature sensors (thermocouples) distributed in several critical heat points of the machine and a sensor measured the ambient temperature. The critical heat points were selected with help of a thermovision system [7]. A temperature state is defined then as the lectures of all temperature sensors in time t_i .

The parametric error, influenced by the thermal deformations, can be written as the combination of two error portions: the parametric error and its variation due to thermal influences which are not the same for each point within the machine working domain. This assumption allows to calibrate each parametric error in two stages. In the first one, the error is read in several points or positions of the machine working plane, just after connecting it, i. e. when it is still cold. The lectures of the parametric error were taken step by step in the forward and backward directions (three measurement cycles). Before reading the parametric error, forwards or backwards, the temperatures indicated for all thermocouples was collected and the temperature gradients calculated. The data set obtained was denominated "data of the initial temperature state". After processing the data obtained, in this stage, with statistical methods [3] the parametric error was expressed as a function from the carriage position (p). The notation $e_{t_0}(p)$ describes the parametric error in the initial temperature state.

In the second stage, the temperature of all thermocouples and the parametric error are collected in several temperature states until the parametric error stabilizes. This information contributes to define the correlation between the error variation and the temperature variation along the carriage travel. The thermal variations can be different for each carriage position, therefore each parametric error should be measured in several positions.

In this work, for this machine, the error variation was observed in four positions, q_i ($i=0, \dots, 3$), along each machine axis, selected after a machine structure analysis. After determining the n positions ($n=4$ for the grinding machine evaluated) the parametric error and the temperature were measured.

The error variation in each position q_i ($i=0, \dots, n-1$) and for each temperature state j can be written as

$$\delta e_j(q_i) = e_{ij}(q_i) - e_{t_0}(q_i) \quad (4)$$

where $\delta e_j(q_i)$ is the error variation in the position q_i and in the temperature state j , $e_{ij}(q_i)$ is the error in the position q_i and in the temperature state j and $e_{t_0}(q_i)$ is the error in the position q_i and in the initial temperature state. $\delta e_j(q_i)$ was calculated for all temperature states evaluated during the calibration process.

In a similar way, the temperature variation for each thermocouples k was calculated, obtaining the vector $\delta_j T$, for all temperature states j . The data set obtained was statistically analyzing with multiple linear

regressions techniques [3] and the variation of the parametric error in each position, q_i , was expressed as a function of the temperature variation. Such expressions were denoted as $H_i(\delta T)$ ($i=0,\dots,n-1$) and they permit to predict the error variation in a position q_i in any temperature state.

In this work, the error variation is describe by a polynomial function of p , measurement position, whose coefficients are functions of the temperature variation δT :

$$\delta e(p, \delta T) = \sum_{i=0}^{n-1} a_i(\delta T) \cdot p^i \quad (5)$$

The coefficients of (5) can be calculated resolving the following linear system:

$$\begin{cases} \delta e(q_0, \delta T) = H_0(\delta T) \\ \delta e(q_1, \delta T) = H_1(\delta T) \\ \vdots \\ \delta e(q_{n-1}, \delta T) = H_{n-1}(\delta T) \end{cases} \quad (6)$$

The parametric error can then be expressed by equation (7):

$$e = e_{i0}(p) + \delta e(p, \delta T) \quad (7)$$

When the parametric errors are substituted in the expressions (2) e (3), the error functions for each preferential direction of the machine are obtained. Then, surfaces showing the Ex and Ez behavior can be traced for different temperature states.

SIMULATED ERROR AND MACHINING ERROR

Verifying the proposed error formulation, a comparison between the error value simulated by the error model developed in this work and the error found in several workpieces, whose diameter were machined by the analyzed grinding machine, was made. The test workpieces were projected with four different diameters. They were first grinded to eliminate the effects of errors originated by previous machining and their diameters were measured with a micrometer.

Three pieces were machined in each temperature state. The diameter error in each temperature state was determinated as the average of the errors measured in the three pieces. During the piece machining the temperature indicated by the thermocouples was monitored. The temperature state is represented by the variation mean value found for each thermocouple. These values were introduced in the expression for Ex. The figure 2 shows the measured piece errors and the simulated errors for several temperature state. In each graphic, the measured error and the simulated error for the four diameter in a determinated temperature state, can be observed. There is an excellent agreement between the predicted and the experimental error. This allows to conclude that it is possible to reduce sensibly the dimensional errors due thermal deformations projecting a error compensation system based on the proposed mathematical model.

CONCLUSIONS

A new formulation for the total error of machine tools considering the parametric errors and its variations due to influences of thermal deformations, was presented in this work. The methodology was applied for modeling the thermal error of a CNC grinding machine and the results were compared with the dimensional errors of workpieces. The results show the excellent model capacity to predict the error propagated to the piece diameter. It is possible to estimate the total machine error for any position and temperature state. By calibration

the data volume was reduced and this made possible the use of commercial software for the data statistical analysis. The calibration method used permits, eventually, observing the error in a position in one preferential direction while the machine works in the other direction, this could be of interest for the machine users. The error equations are relative simple and could be used for other types of machine tools and for the implementation of its error compensation systems.

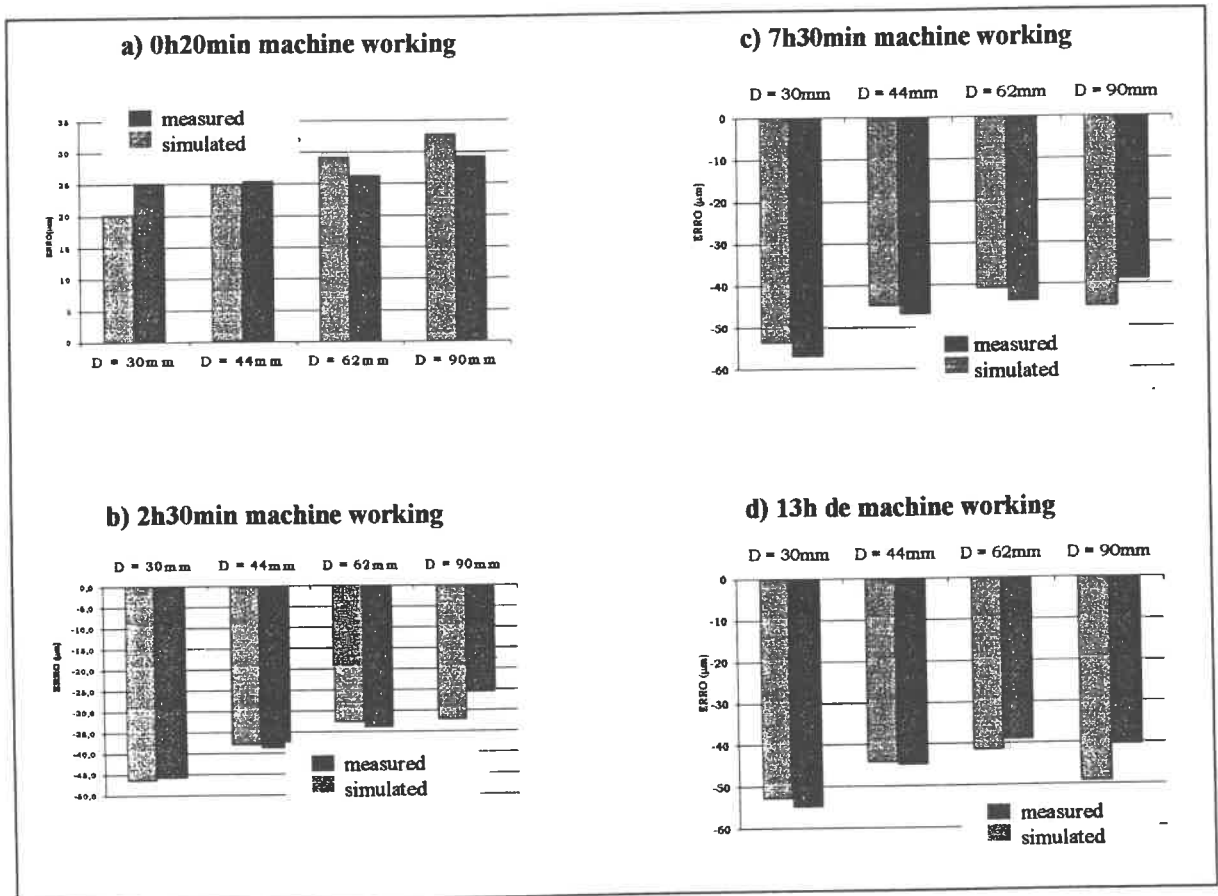


Figure 2: Measured and simulated errors.

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OPTIMIZATION-BASED CONCEPT DESIGN OF A MACHINING CENTER

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Abstract

Stiffness is one of the major considerations in the design of a machine tool. It is largely established at the stage of conceptual design when the major dimensions and the joints are determined. In this paper an optimization process conducted at the conceptual design stage to improve the structure of a machine tool is presented. As a result of optimization the objective function involving the cost of the joints and the weight of the machine is effectively reduced while the required stiffness is maintained.

1. Introduction

In the process of designing a machine tool, the finite element method (FEM) is often used to simulate the stiffness of the machine against loads. In most design practices, if the stiffness from FEM analysis is not satisfactory, design variables are modified manually to improve stiffness. To improve design with minimum manual work the application of optimization is considered. Theoretically, an optimization process can be linked to the FEM analysis as a guide to how the design variables should be modified to improve stiffness. However, this approach may not be very effective. FEM is often applied at the detail design stage when most of the detail dimensions are already defined. If optimization is applied at this stage modifications in design may be restricted to local areas and improvements to stiffness may be very limited.

In fact, the principal design variables including the major dimensions and the selection of joints have dramatic and global effects on the stiffness of the machine. Therefore, a better way to improve the stiffness effectively is to apply optimization early at the stage of concept design when the principal dimensions are determined.

In this paper, an approach to improve the structural design of a machine tool at the concept design stage is introduced. A vertical machining center is used as an example to demonstrate the process and the effects of optimization. In phase I of this research the structural parts of the machining center are assumed to be rigid while the joints at the interfaces of structural parts are the sources of deformations of the machining center under loading. The selection of the joints and some major dimensions of the structure are considered the design variables in the optimization process. In the following sections the method of displacement calculation and that of optimization are introduced. The results before and after optimization are then presented and discussed.

2. Calculation of Displacements

There are five structural parts in a vertical machining center: head, column, table, saddle and base, as shown in Figure 1. The joints under consideration are those located at the interfaces between the head and the column, the table and the saddle as well as the saddle and the base. The set of joints at each interface includes a feed screw system and a set of linear rolling guides. Because the structural parts are assumed to be rigid in this phase of the research, the displacement between the tip of the spindle and the top of the work piece is mainly due to deformation at the joints.

The coordinates used in the analysis are defined in Figure 1. An external load is applied at the tip of the spindle and transmitted to the interface between the head and the column. Meanwhile, reaction force with the same magnitude but in the opposite direction is applied at the top of the work

piece. It is transmitted to the interface between the table and the saddle as well as to the interface between the saddle and the base.

At the interface between the head and the column there are four ball slides attached to the plate at the rear of the machine tool head. To evaluate the displacement at the spindle tip (d_s), the first step is to calculate the force and the moment at the center of the plate transmitted from the spindle tip. The force and the moment are then converted into equivalent forces applied to the four ball slides and the feed screw system. The equivalent forces have components in three directions: normal to the guides (Y direction), lateral to the guides (X direction) and axial of the feed screw shaft (Z direction). The normal force is supported by the downward/upward stiffness of the guides, the lateral load is supported by the lateral stiffness of the guides, while the axial load is taken by the axial stiffness of the ball screw system.

For linear rolling guides, the stiffness against loads in downward/upward and lateral directions subject to various preloads can be found in manufacturers' catalogs.¹ For the feed screw systems, the axial stiffness K_T is a combination of the axial stiffness of the screw shaft, the nut, the supporting bearing, and the nut and bearing mounting section.¹ Once the stiffness of the linear rolling guides and the feed screw system are obtained, the deformation in x , y , z , θ_x , θ_y , θ_z at the head-column interface can be evaluated. Then, with the location of the spindle tip relative to the interface available, the displacement at the spindle tip d_s can be calculated.

The displacement at the top of the work piece, d_w , due to the deformation of the table-saddle interface and the saddle-base interface can be obtained in a similar way. The sum of the absolute values of d_w and d_s indicates the effect of the stiffness of the joints at the three interfaces and can be included as part of the objective and/or constraint functions in the optimization program.

3. Method of Optimization

The design variables in the optimization program are the choice of joints and some dimensions involved in stiffness calculation. The joints, represented by the serial number of the feed screw parts and that of the linear rolling guides, are integer variables. The dimensions involved in stiffness calculation are the dimensions displayed in Figure 2, including Ry , Rz , Dc , Lc , Ds , Ls , Db , and Lb . Unlike the choice of joints, the above dimensions are real numbers.

The Objective function in the optimization program includes the cost of the joints and the weight of the structure parts. However, because the weight of the structure is not available at the current stage, it is substituted temporarily by a simple function of Ry , Rz , Dc , Lc , Ds , Ls , Db , and Lb . More accurate calculations for the weight of the structural parts will be included in the next phase of the research.

The constraint set includes:

- * The bounds of design variables, that is, the upper and the lower bounds for the serial numbers representing the choice of joints and those for the dimensions involved in the stiffness calculation.
- * The upper bounds of d_i , the displacements between the tip of the spindle and the top of the work piece. Four constraints are included: the upper bounds of the individual components of d_i in x , y , z directions and that of d_i itself

Because nonlinear functions and integer variables are involved in the objective and the constraints, the GRG (Generalized Reduced Gradient) method developed by Lasdon and Waren² combined with the Branch and Bound method is applied in the solution procedure. The optimization software used in the project is Premium Solver Plus.³

4. Results and Discussion

The results of optimization are summarized in Table 1. The objective function was reduced

from 6047.95, the initial value, to 3364.50 following optimization. As mentioned before, the objective function is a combination of the cost of the joints and the weight of the structure parts and thus is a performance index without unit. The initial values and the optimized ones for the design variables are also listed in Table 1.

Items	Initial Values	Optimized Values
Objective function	6047.95	3364.50
Ry	530	500
Rz	401	391.31
Dc	362	304.78
Lc	340	300
LRG*(head/column interface)	10	3
FSS**(head/column interface)	8	3
Ds	319	250
Ls	576	350
LRG* (table/saddle interface)	10	3
FSS** (table/saddle interface)	8	4
Db	545	400
Lb	351	250
LRG* (saddle/base interface)	10	3
FSS** (saddle/base interface)	8	4

* LRG: Linear Rolling Guides

** FSS: Feed Screw System

Table 1 Results

In the results presented in Table 1, the structural parts are assumed to be rigid. To improve further the design of the machine tool, the weight and the deflections of the structural parts should be estimated and included in the optimization process. A method to estimate the effects of the principal variables on structural deflection and weight is currently under investigation. The effects of the principal variables are represented as mathematical functions and included as part of the objective and constraint functions in the optimization process to determine the best values for the major dimensions, the joints and the types of structural reinforcement.

5. Conclusion

In this paper an optimization process conducted at the conceptual design stage to improve the design of the principal dimensions and the selection of the joints is presented. As a result of optimization the objective function involving the cost of the joints and the weight is effectively reduced while the deflections are constrained to be less than the values assigned by the user. Future work includes the development of a method to estimate the deflections and weights of the structural parts and include them in the optimization process.

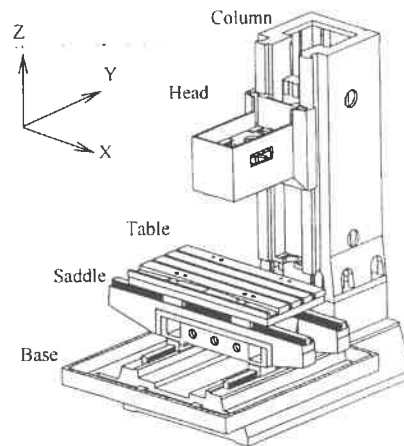


Figure 1 A vertical machining center

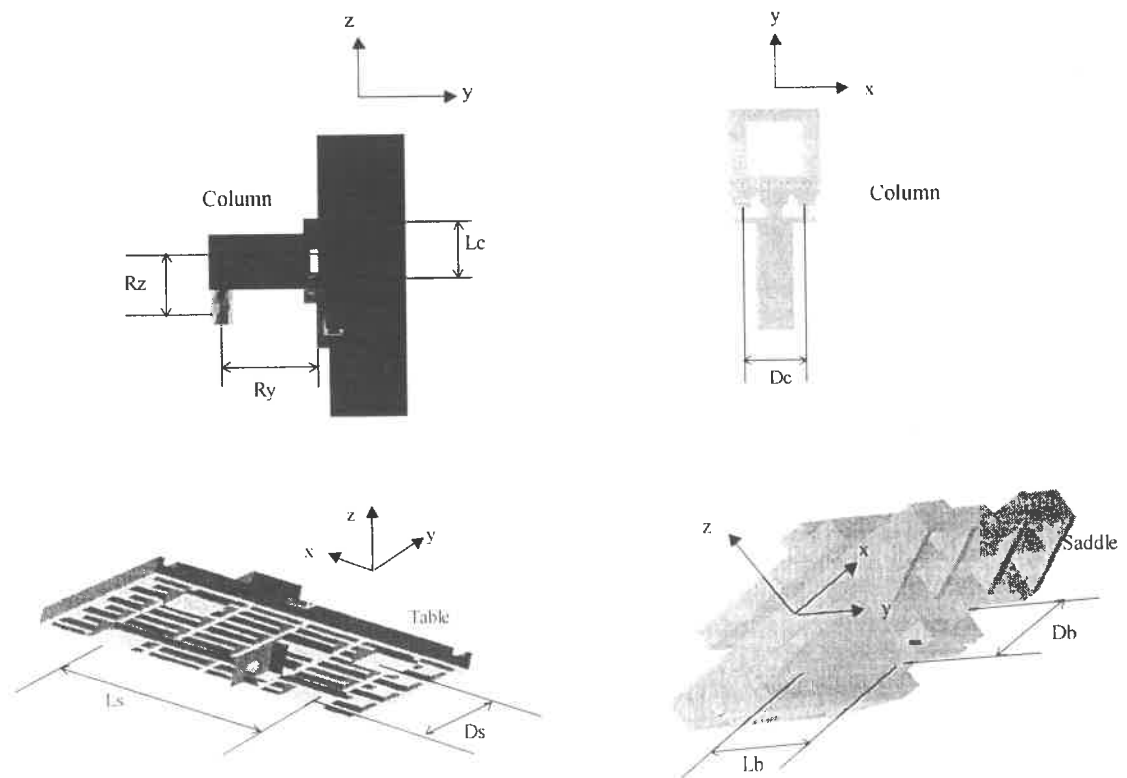


Figure 2 Dimensions involved in the stiffness calculation

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HIGH PRECISION MEASUREMENT OF TORQUE IN STIRRED VESSELS

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ABSTRACT

An optical system for the measurement of torque in small stirred vessels was developed. Torsion stress of the shaft was determined using a laser inciding a photoelastic material. A He-Ne polarized laser beam inciding on an acrylic coated shaft is analysed using a quarter wave plate and detected with a silicon photodetector to sense the torsion stress in the shaft produced in a controlled way. This technique permit us to determine low torque values accurately. Static calibration was carried out by using a lever arm coupled to the shaft, in which different weights were placed at a certain distance from the shaft axis. Preliminary results of this calibration are presented.

INTRODUCTION

Mixing is perhaps one of the most common processes found in the processing industries (i.e. chemical, food, pharmaceutical, etc.). From an engineering point of view, power consumption is perhaps the most important design parameter. The measurement of the energy input, P , is of major importance for the design and operation of mixing equipment, but the costs associated with the power input contribute importantly to the overall costs of industrial plants.

The knowledge of the power input, in a specific stirred tank reactor, is necessary for scale-up and scale-down procedures, process characterization, modelling, mixing performance as well as heat transfer and fluid dynamics studies. The success of a fermentation process depends strongly on the energy transmitted by the impellers to the liquid in the bioreactor tank, therefore the power input is an important factor in the determination of process costs.

A number of methods are available for the measurement of the power input in agitated vessels. In large scales, wattmeters and ampermeters are suitable. In pilot scale, dynamometers coupled to the shaft are commonly used. In small laboratory scale, high precision measurements are difficult to perform. Optical technique has been used for the measurement of torque in a milling machine [4], and this appears to be a good option for determining small values of torque in agitation shafts, particularly in small vessels, in which other methods (i.e. strain gauges) are impractical. For instance, in small fermentors.

When a photoelastic material is subjected to a load, its structure is oriented in the general direction of the applied load. If a polarized light beam incides on a dielectric material, a beam irradiance fraction is reflected. The variation of this fraction depends on the reflexion angle as well as on the polarization component associated to the light beam [3].

When photoelastics (i.e. acrylic) are deformed, the long-chain molecules align in the direction of the applied load. This process is called orientation and, as in metals, the polymer becomes anisotropic [2].

In our case the incident light beam will experience an anysotropic medium, which produces a variation of the reflected beam in terms of the polarization with respect to the incident beam. This variation depends basically on the magnitude and direction of the applied load.

EXPERIMENTAL SET UP

The system used in this work is shown in figure 1. It consisted of a He-Ne laser with a wavelength of 638.2 nm, a $\lambda/4$ wave plate, a beam splitter, a silicon photodiode, a linear polarizer, a positive lens with a focal distance of 100 mm and a four channel oscilloscope used as the detection and signal processing system. A brass shaft with a 0,187 inch diameter and 4 inch lenght with an acrylic coating was used for the experiment. In the optical system a laser beam passes through a polarizer and then is transmitted through a $\lambda/4$ retarder wave plate and is focused with a positive lens and is inciding to the photoelastic coated beam shaft. The reflected laser beam passes through beam splitter and goes to photodiode and then to an amplifier where the signal is amplified by a factor of 10^2 and then goes to the four channel oscilloscope.

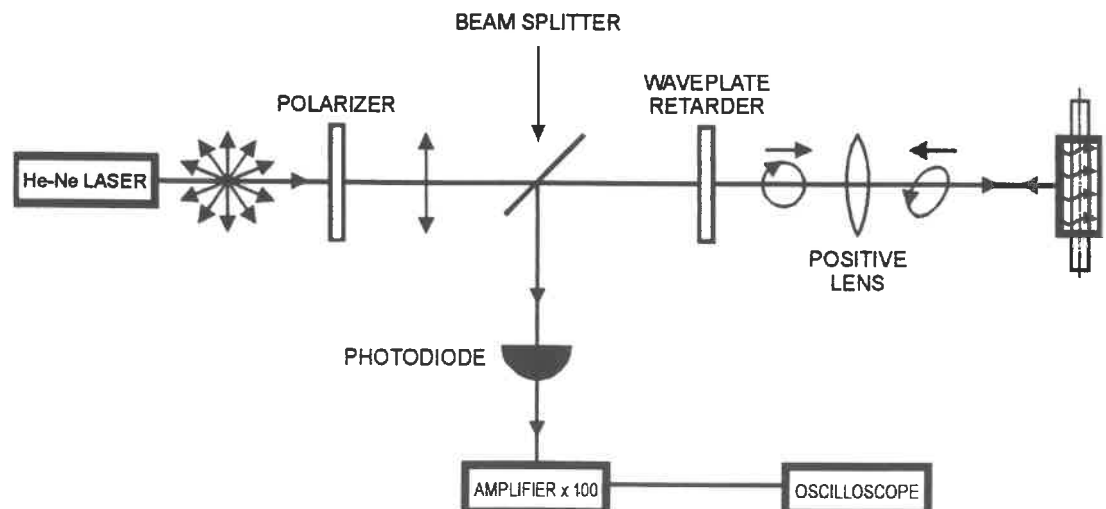


Figure 1 Optical system for measuring the torque in a photoelastic coated shaft

CALIBRATION

The static calibration of the system was carried out by using an acrylic coated shaft supported by a metallic structure. In the shaft, a torsional stress was applied by means of different weights from 20 to 500 g placed at 0.16 from the shaft axis as shown in figure 2.

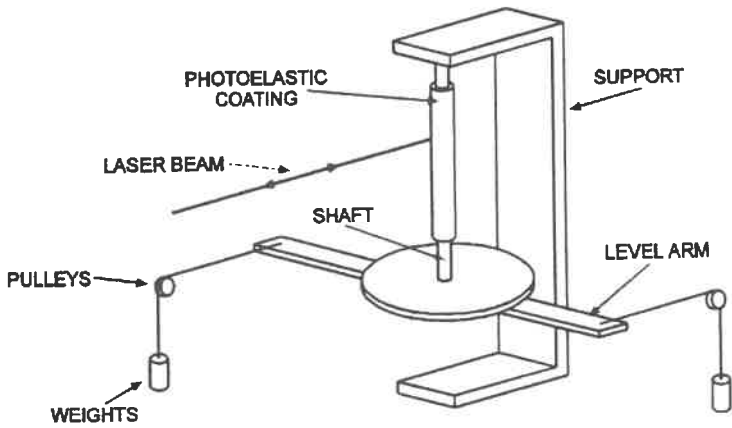


Figure 2 Static Calibration

A 100 factor was used in the amplifier to measure the signal in the oscilloscope. As it shows the curve calibration of figure 3, a straight line was obtained, having a slope of 4.898 Nm/V.

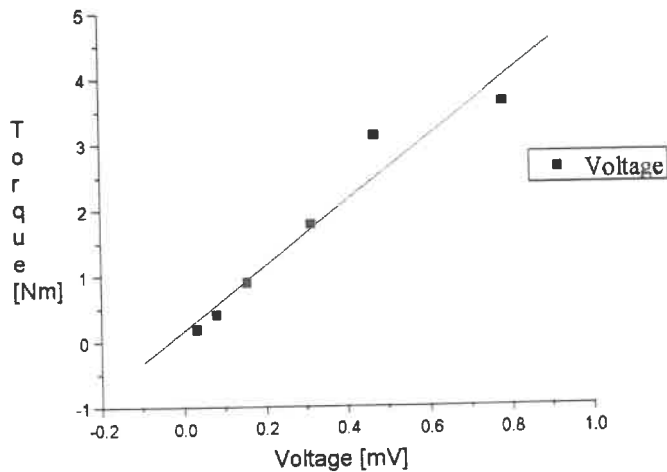


Figure 3 Calibration curve

CONCLUSIONS

Results of static calibration showed that the optical system promising for measuring accurately small values of mixing torque. A detection and signal processing will be developed for making the measurement simpler and with no need of the oscilloscope. Having the calibration obtained with the oscilloscope, small values of power drawn in stirred vessels can be determined. Testing of the system under dynamical conditions in small laboratory fermentors is in progress.

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Position Measurement on Machine Tools: By Linear Encoder or Ballscrew and Rotary Encoder?

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a discussion as to whether linear encoders or rotary encoders and recirculating ballscrews represent the best solution for measurement on NC machine tools is experiencing a renaissance at present. This is to some degree certainly an aftereffect of the recession in the machine tool industry, but it also springs from recent changes in drive technology.

After the general recession in the machine tool industry, in several quarters the question arose of whether the industry's strategy of the last few years has been correct. Could it be that too little attention is paid to cost and too much to pushing the limits of the technologically possible? Is it not preferable to move those features not required for a basic version of a machine tool, permit a competitive starting price for a low-end model? Under the pressure of such currents of opinion one can easily come upon the idea of saving money on the measuring systems by leaving linear encoders on the lowest-priced models and offering them as an accuracy-enhancing option.

End toward digitally driven axes
This line of thought is reinforced by the trend in the drive industry in the

direction of digital axes. A large portion of the new servo motors now feature rotary encoders, which in principle can also be used in combination with the ballscrew for position control. With this method there is no longer a question of choosing between a ballscrew/rotary encoder system and a linear encoder solution for position control. With such a drive configuration the decision is rather whether to add a linear encoder or simply to use an already existing motor encoder working in combination with the ballscrew.

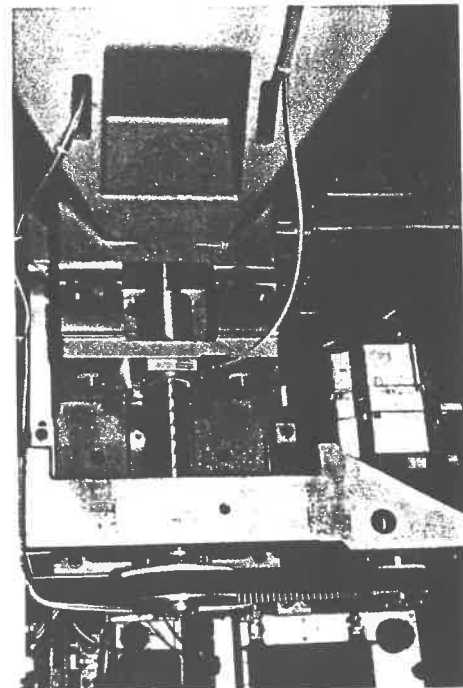
In the heat of such discussions one should not forget to consider the problems known to be involved with position measurement using a rotary encoder/ballscrew system.

They can quickly increase the cost of an „economical“ machine if the owner finds that the accuracy does not suffice in certain applications, or that thermal expansion problems are causing the machine to generate scrap every morning. Retrofitting an installed machine with optional linear encoders is usually much more expensive than having them delivered with the machine from the start.

An exact error analysis of position measurement via rotary encoder and ballscrew begins with a consideration of prevalent mechanical feed-drive systems.

Mechanical feed-drive system

Although machine tool designs vary immensely, the mechanical configuration of their feed drive is largely standardized (**Fig. 1**), (photograph). In almost all cases, the recirculating ballscrew has established itself as the solution for converting the rotary motion of the servo motor into linear slide motion. The ballscrew is normally fixed in axial direction at only one end with preloaded angular-contact ball bearings, which take up the axial forces of the slide. The servomotor and ballscrew drive can be connected by a toothed-belt trans-



Feed axis drive on milling machine

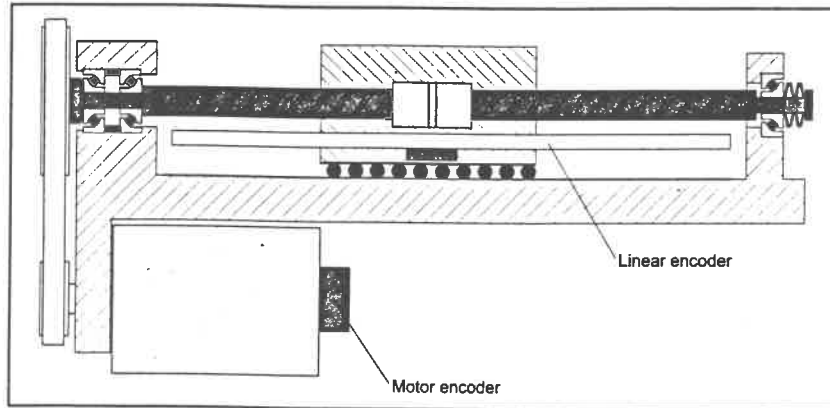


Fig. 1: Typical drive system of a numerically controlled machine tool with linear scale on the slide and rotary encoder on the motor.

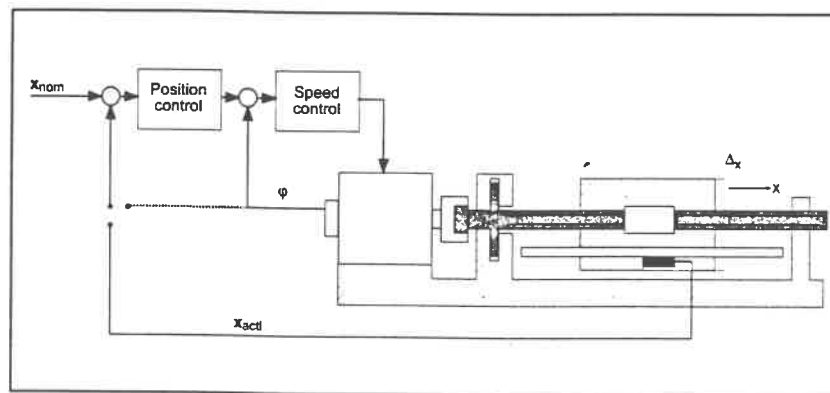


Fig. 2: Fundamental difference between position control with linear encoder and with rotary encoder/spindle. The linear encoder includes the feed drive mechanism in the control loop.

mission to achieve a compact design and better adapt the speed. Direct coupling configurations are often found in non-European machines.

A position control loop via rotary encoder and ballscrew includes only the servomotor (Fig. 2). In other words, there is no actual position control of the slide, because only the position of the servomotor rotor is being controlled.

To be able to extrapolate the slide position, the mechanical system between the servomotor and the slide must have a known and, above all, reproducible mechanical transfer behavior.

A position control loop with a linear encoder, on the other hand, includes the entire mechanical feed-drive

system. The linear encoder on the slide detects mechanical transmission errors and these are compensated by the machine control unit.

Differing terminology

Differing terms are used to distinguish between these two methods of position control. German-speaking and some English-speaking communities generally refer to them somewhat inaccurately as „direct and indirect measurement.“

Here the Japanese concepts of „semi-closed-loop and closed-loop control“ seem appropriate, since they more aptly describe the actual problem.

Kinematic error

Kinematic error in position measurement with rotary encoder and ball-screw results primarily from ball-screw pitch error. This error directly influences the result of measurement because the pitch of the ball-screw is being used as a standard for linear measurement.

Reversal error

Reversal error occurs during positioning from differing directions. The causes are play and elasticity in connection with frictional forces.

But also the so-called pitch loss [1] resulting from a shift of the balls during the positioning of ballscrew drives with two-point preloading can lead to reversal error in the magnitude of 1 to 10 μm .

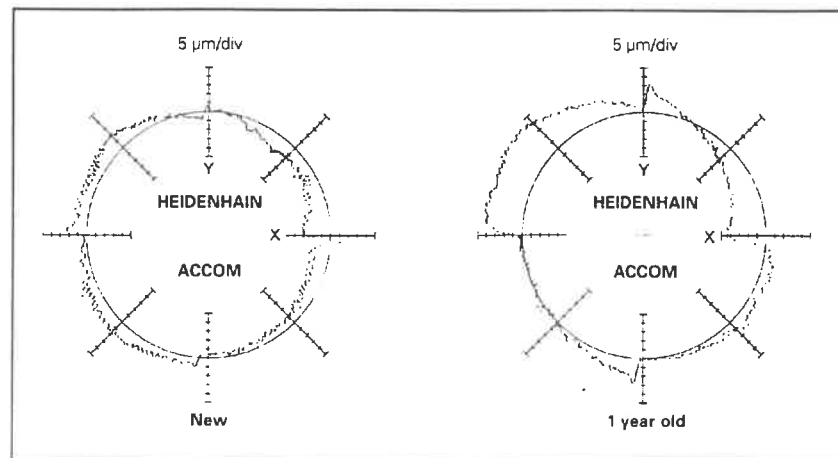


Fig. 3: Circular tests of a machining center without linear encoders in new condition and after one year. The reversal error has significantly increased in the X axis.

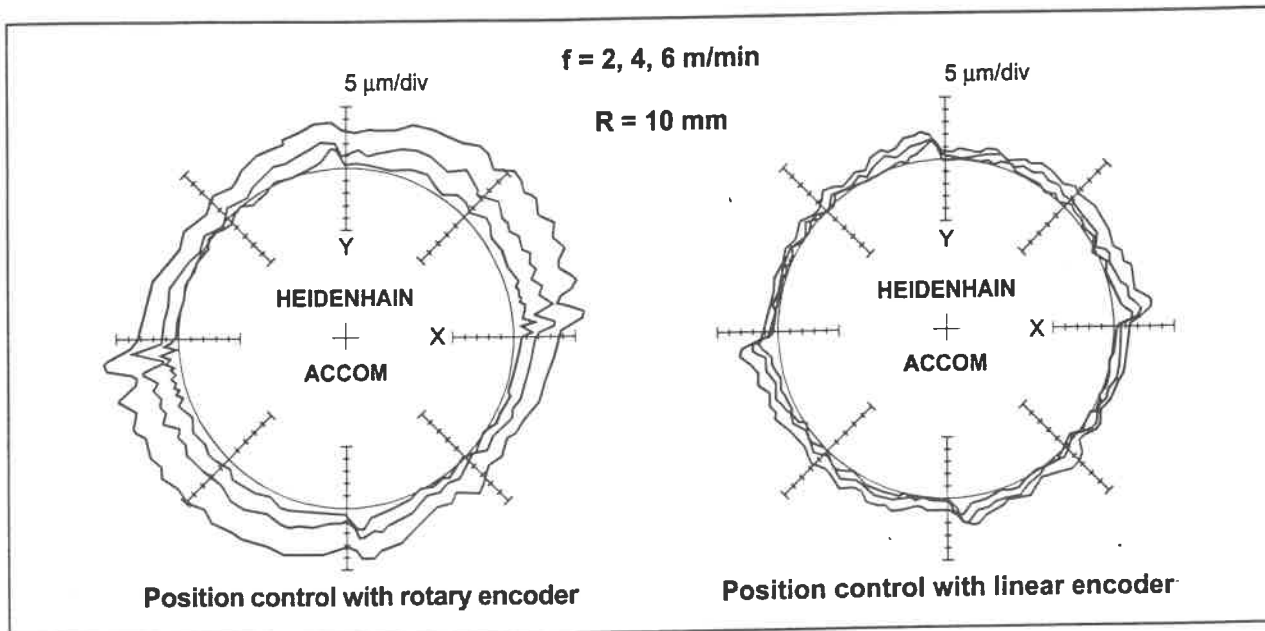


Bild 4: Circular tests of a machining center that has been retrofitted with linear encoders. With position control via rotary encoder and spindle, the circle deviates significantly from the ideal path at high velocity. With linear encoders the contour accuracy is significantly better.

Error compensation

Most controls are capable of compensating pitch error and reversal error. However, to determine the compensation values it is necessary to make elaborate measurements with comparative measuring devices such as interferometers and grid encoders.

In addition, the reversal error is often unstable over long periods of time and must be regularly recalibrated (Fig. 3). The causes of this instability include run-in processes of the ball-screws and changes of the frictional forces in the guideways. Toothed belt drives can also cause significant positioning errors in the course of time.

Deformation of drive mechanisms

Forces leading to the deformation of feed drive mechanisms are essentially inertia forces resulting from acceleration of the slide, cutting process forces, and friction in the guideways. They cause a shift in the actual axis slide position relative to the position measured with the ball-screw and rotary encoder.

Forces of acceleration

The mean axial rigidity of a feed drive mechanism as shown in Figure 1 lies in the range of 100 to 200 N/ μm (with a distance between ball nut and fixed bearing of 0.5 m and a ball-screw diameter of 40 mm). A typical slide mass of 500 kg and a moderate

acceleration of 2 m/s^2 result in deformations of 5 to $10 \mu\text{m}$ that cannot be recognized by the rotary encoder/ballscrew system. The present industry trend toward accelerations in significantly higher ranges will result in increasingly great deformation values.

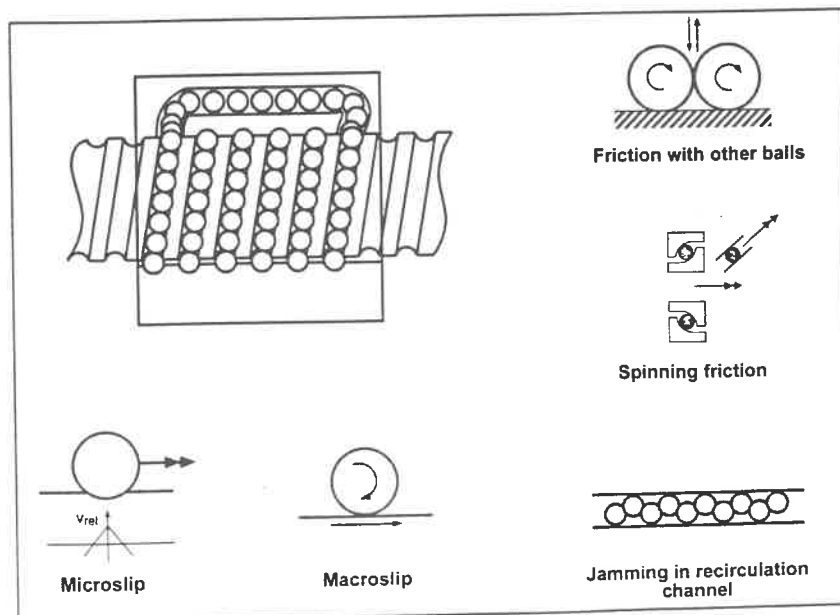


Fig. 5: Causes of the high moment of friction in preloaded ballscrews

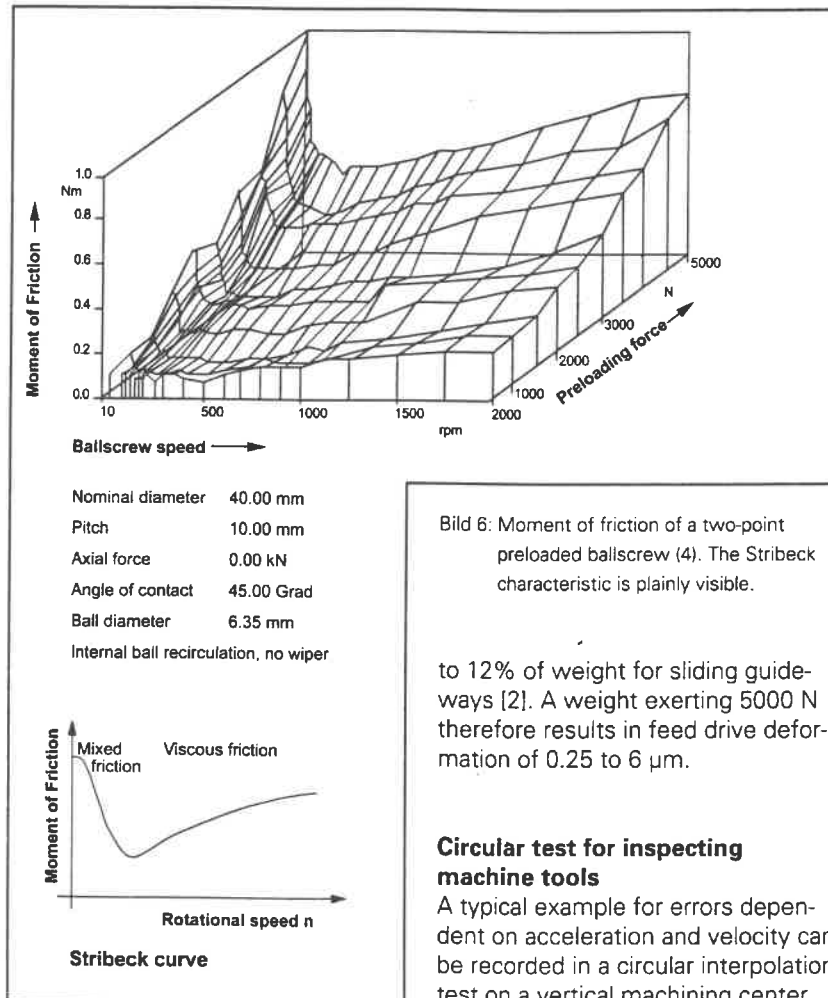


Bild 6: Moment of friction of a two-point preloaded ballscrew (4). The Stribeck characteristic is plainly visible.

to 12% of weight for sliding guide-ways [2]. A weight exerting 5000 N therefore results in feed drive deformation of 0.25 to 6 μ m.

Circular test for inspecting machine tools

A typical example for errors dependent on acceleration and velocity can be recorded in a circular interpolation test on a vertical machining center (Fig. 4). Where position control is by rotary encoder and ballscrew, the circles traversed at higher velocities deviate significantly from the ideal path. The same machining center shows significantly better contour accuracy when equipped with linear encoders.

Cutting forces

The cutting forces can quite possibly lie in the kN range, but their effect is distributed not only in the feed drive system, but also over the entire structure of the machine between the workpiece and the tool. The deformation of the feed drive system therefore normally has only a small share in the total deformation of the machine. A linear encoder can recognize and correct only this small portion of the total deformation. Critical component dimensions, however, are normally finished at low feed rates, so that the deformation of the feed drive system is negligible.

Forces of friction

The forces of friction in the guide-ways lie between 1% and 2% of weight for roller guideways and 3%

Positioning error due to ballscrew expansion

Positioning error resulting from thermal expansion of the ball screw presents the greatest problem for position measurement via rotary encoder and ballscrew. This is because the ballscrew drive must serve a double function: On the one hand it must be as rigid as possible to convert the rotary motion of the servo motor to linear feed motion. On the other hand it must serve as a precision measuring standard. The two-fold function therefore forces a compromise because both the rigidity and the thermal expansion depend on the preloading of the ball nut and the fixed bearing. Both the axial rigidity and the moment of friction are roughly proportional to the preloading.

Friction in the ball nut

The largest portion of the friction in a feed drive system is generated in the ball nut. This is because of the complex kinematics of a recirculating ball nut. Although at first glance the balls may seem only to be rolling, they are in fact subjected to a great deal of friction. Besides the microslip resulting from relative motion in the compressed contact areas, the greatest effect is from the macroslip due to kinematic exigencies.

The balls are not completely held in the races and wobble much like tennis balls rolling down a gutter. The result is a continual pressing and pushing with occasional slipping of

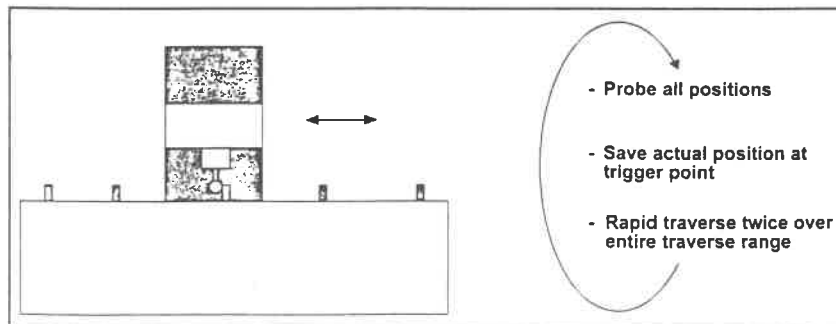


Bild 7: Simple experimental setup showing thermal growth in a recirculating ballscrew. With a touch trigger probe, fixed positions within the traverse range are probed and the respective actual position values of the linear encoder and of the rotary encoder/spindle system are recorded.

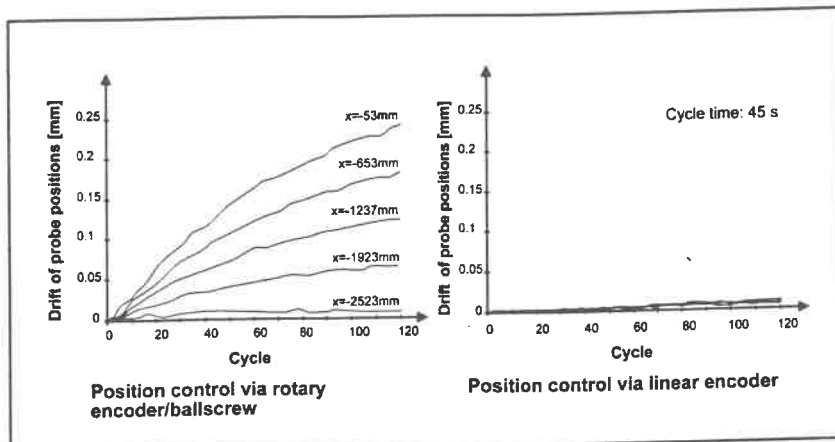


Fig. 8: Results of the experiment shown in Figure 7. With position control using rotary encoder/spindle there is a significant drift in the probe position values as a result of thermal expansion in the ballscrew.

the balls. The friction among the balls is aggravated by high surface pressure due to the absence of a retaining device to separate them. As in every angular-contact ball bearing a spinning friction results from a contact diameter that is not orthogonal to the axis of ball rotation. Each ball therefore rotates about its contact diameter. Recent studies have also shown that the balls can move in the thread only because of an additional slip component brought on by the thread pitch [3].

The recirculation system is a special problem zone for ball screws. With every entrance into the recirculation channel, just as with every exit, the movement of the ball changes entirely. The rotational energy of the balls, which in rapid traverse typically rotate with 8000 rpm, must be respectively started and stopped. In contrast to the preloaded thread zone, in the recirculation zone the balls are not under stress. For reasons of

energy the balls tend to collect in the recirculation channel. Without elaborate measures to reintroduce the balls into the thread at the end of the channel it tends to congest, causing the familiar jamming of the ball screw drive.

Frictional heat generated in the ball nut

The moment of friction of a ground precision recirculating ballscrew with 40 mm diameter and 10 mm pitch was measured by Golz [4] for various preload forces and rotational speeds (Fig. 6). The Stribeck characteristic of frictional moment is clearly recognizable. It confirms the high share of solid-body friction and mixed friction in ballscrew drives at low speeds. Viscous friction dominates at high speeds. It is interesting to note that for this typical ballscrew the normal machining feed rates lie far below the speeds at which the moment of friction is at its minimum.

The rapid traverse feed rates, however, lie far above it. The feed rates at which this ballscrew is at optimum efficiency therefore seldom occur. The moment of friction is only slightly dependent on the axis load of the ball nut [4]. With a typical preload of 3 kN and allowing for the missing wiper, this results in a no-load or frictional moment of 0.5 to 1 Nm. This means that in rapid traverse at a ballscrew speed of 2000 rpm approximately 100 to 200 W of frictional heat is generated in the ball nut.

Drift of position values measured with rotary encoder and spindle

A simple experiment (Fig. 7) shows the influence of frictional heat on the positioning behavior of a machining center. On a large machining center equipped with linear encoders and rotary encoders, five positions in the X axis were repeatedly measured with a touch probe, and the respective actual values of the rotary encoder and linear encoder at the touch points were recorded. After all five positions were probed, the axis was moved over the entire traverse range in two cycles at rapid traverse. In total, 120 of these cycles were performed within 90 minutes.

Warm-up phase

The drift of the individual probe positions relative to their starting value plainly shows the thermal growth of the ballscrew (Fig. 8). When the positions are measured with rotary encoder and spindle, the position farthest from the fixed bearing drifts by approx. 250 μm . It is interesting to note that the drift increases very

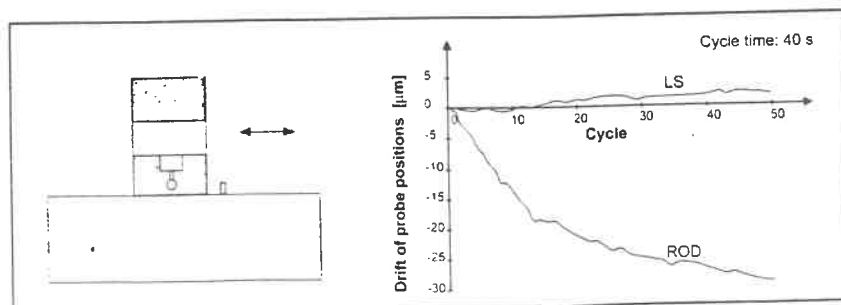


Bild 9: Typical drift of the actual position values during a work break. The actual position values - measured by rotary encoder and spindle - drift by approx. 30 μm within 30 minutes.

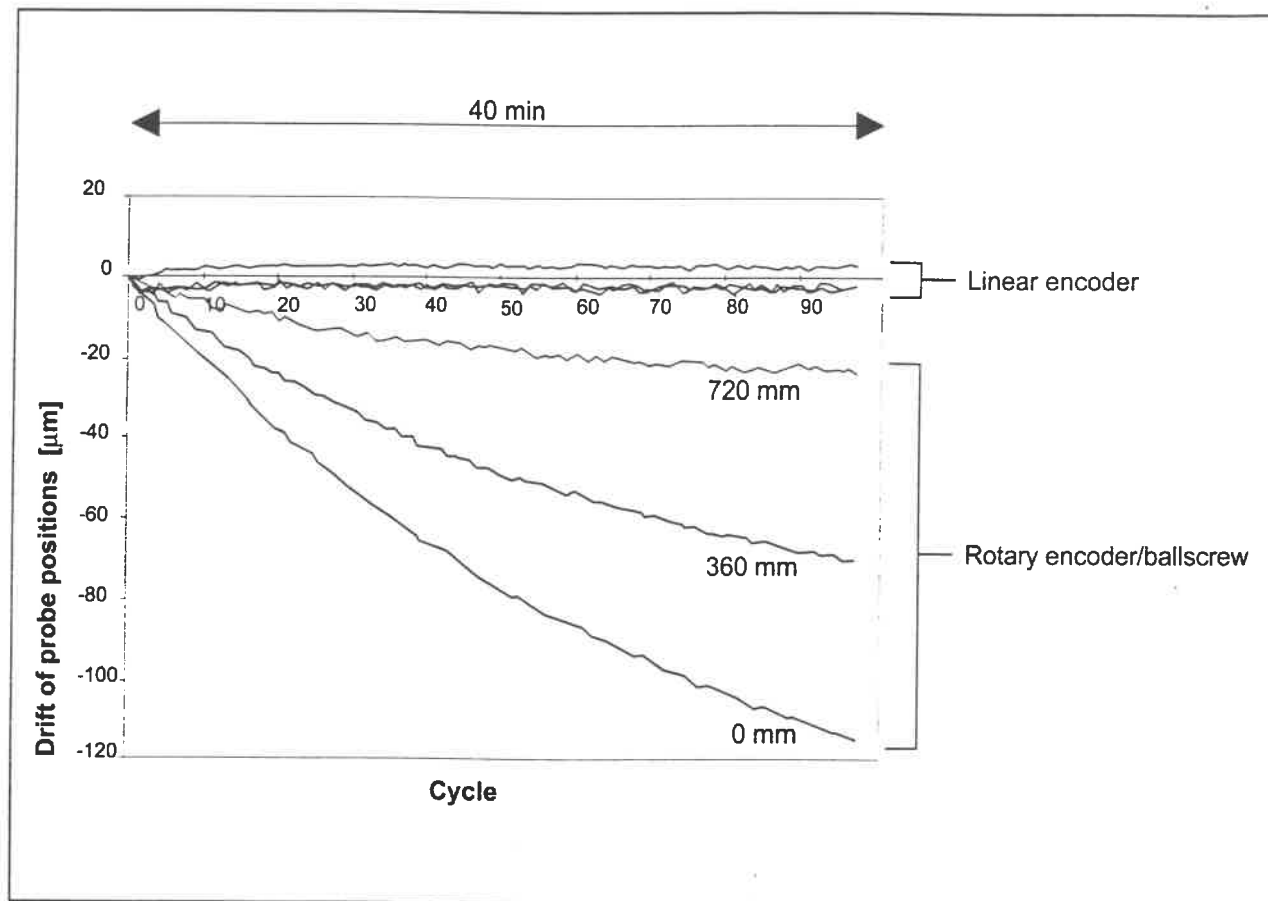


Fig. 10: Drift of three positions during measurement in accordance with ISO 230 with a large number of repetitions. Position measurement via rotary encoder and ballscrew shows a significant drift of positions due to the thermal growth of the ballscrew.

quickly immediately after switch-on. Any change in the mean feed rate during a machining operation therefore immediately affects the positioning accuracy. Similar results were published by Schmitt [5].

Cool-down phase

Figure 9 shows the opposite effect. The same machining center was stopped during series production of small steel parts, and one work position was repeatedly probed. The drift in the position measured with rotary encoder and ballscrew of 30 μm over 30 minutes is again clearly visible.

Measurement of position accuracy according to ISO 230

The thermal expansion of the ballscrew

can also be shown when positioning accuracy is measured repeatedly according to ISO/CD230-3. **Fig. 10** shows the position values of such a unidirectional measurement relative to their initial value on a vertical machining center retrofitted with linear encoders. The 1-m long X axis was moved to three positions at 10 m/min a total of 100 times. At first, position control was through the rotary encoder/ballscrew system. In a second experiment under otherwise identical conditions, position control was through the linear encoders. The comparator measuring device was a VM 101 manufactured by HEIDENHAIN.

No drift in position values measured with linear encoders

In spite of the moderate feed rate of

10 m/min (rapid traverse 24 m/min, the position farthest from the ballscrew fixed bearing drifted by more than 110 μm within 40 minutes. A significant growth of the ballscrew becomes apparent already after the first few cycles. The measured positioning accuracy therefore depends directly on the number of repetitions, particularly after the first few repetitions. The measurements taken by the retrofitted linear encoder show no drift.

Batch production

The following experiment shows the influence of thermal growth in batch production with fixed clamping positions. Eight 70 mm x 70 mm workpieces were fixed on a vertical machining center (**Fig. 11**). Four pockets and two radii were machi-

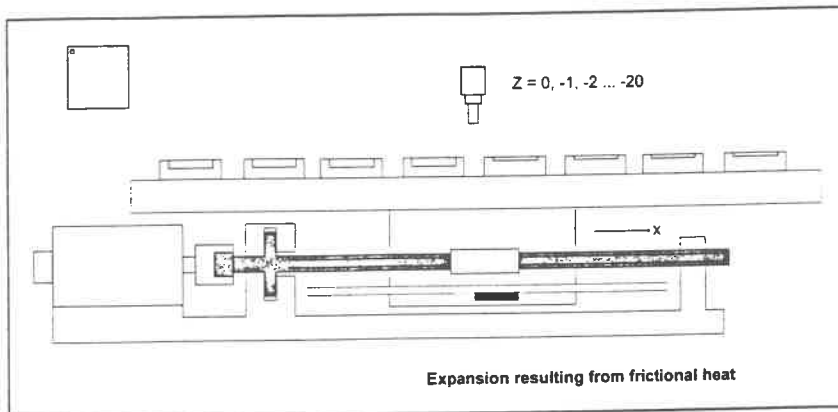


Fig. 11: Experimental setup for batch production with multiple workpieces. To illustrate drift resulting from thermal expansion of the ballscrew, the workpieces were not exchanged after machining. Instead, the part program was run repeatedly at successively increasing depth.

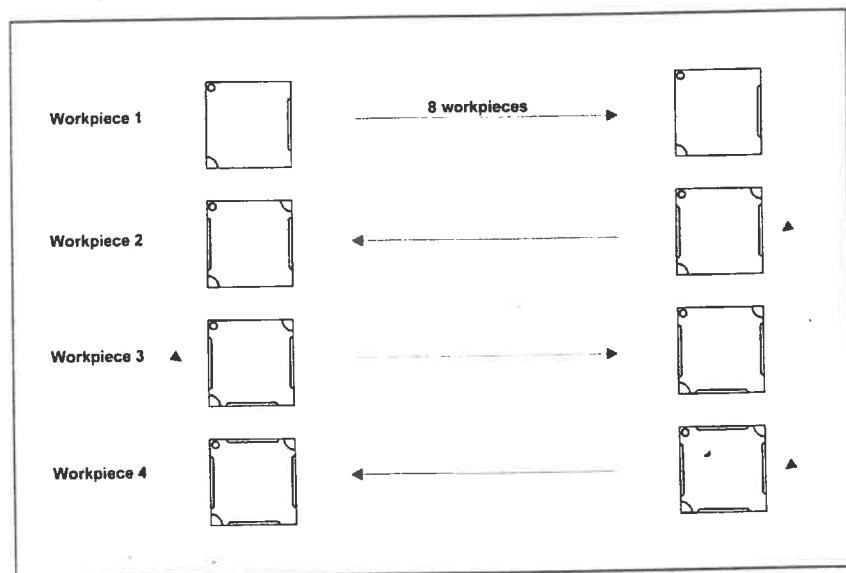


Fig. 12: Part program used in the experiment shown in Figure 11. Four pockets and two radii were machined using 4 tools at an infeed of 1 mm.

ned using 4 tools with an infeed of 1 mm in the Z axis (**Fig. 12**). After the 6-minute machining operation the 8 parts were not exchanged. Rather, the infeed in Z was increased by 1 mm and the operation repeated. As a result of the thermal expansion of the ballscrew, all workpieces show a step pattern on the left side. This pattern is particularly pronounced on the workpiece farthest to the left. The right sides of the workpieces are smooth because with each shift in the positive X direction the previous step was also removed. In principle the same effect could be observed in the Y direction as in the

X direction, but because of the lesser amount of movement in the Y axis the step pattern is significantly less pronounced. In the X direction the comparative measurement of the step pattern shows a drift of approx. 90 μm with a time constant of thermal expansion slightly less than an hour.

If additional work is to be done on previously machined workpieces with critical dimensions, the machine datum must be continually inspected and corrected. The machine achieves thermal equilibrium after one hour, but after an interruption in

machining it begins to drift in the reverse direction. If the part program and with it the mean feed rate are changed, it again takes approx. 1 hour for the ballscrew to regain thermal equilibrium.

The same experiment was conducted with linear encoders. The results of machining showed virtually no drift (**Fig. 13**).

Various measures are presently being discussed and realized to counter positioning error resulting from ballscrew expansion.

Some manufacturers offer hollow ballscrews that conduct coolant to prevent thermal expansion.

Circulation of the coolant through rotating ballscrews requires rotary leadthroughs near the ballscrew bearings. Besides the sealing problems obviously involved, this method presupposes the availability of a temperature-controlled coolant, which is usually not the case. Also, it reduces the rigidity of the ballscrew in the direction of traverse. Presumably, the cost of this method is greater than the cost of linear encoders.

Many studies are presently being conducted on the compensation of thermal deformation with the aid of analytical models, neuronal networks and empirical equations. In most cases, however, these studies focus on thermal expansion caused by main spindles.

To compensate the expansion of ballscrews, the temperature must be known as a function of position, because in some part programs thermal expansion can occur locally. Direct temperature measurement of the rotating ballscrew, however, is very costly. The possibilities of drawing conclusions on ballscrew expansion using temperature data from the ball nut or bearing are very limited.

The application of fixed bearings at both ends of the ballscrew does sig-

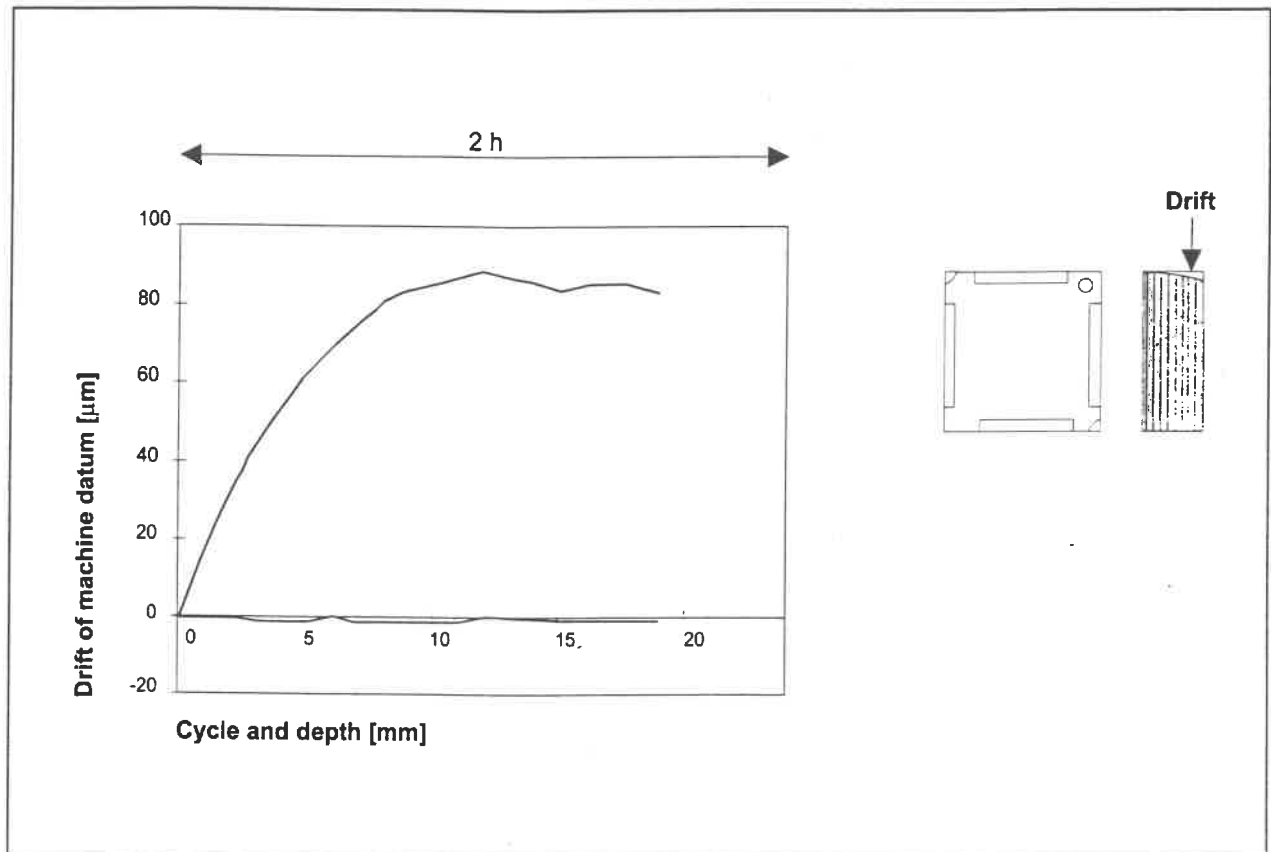


Fig. 13: Result of the experiment in Figure 11. The left pocket of the left workpiece plainly shows a step pattern resulting from thermal expansion of the ballscrew.

nificantly increase the axial rigidity of the feed drive, however it can hardly prevent thermal growth of the ballscrew. For a ballscrew diameter of 40 mm, the bearings would have to apply approx. 2.6 kN/K of force to suppress the thermal expansion. For the typical temperature increase of more than 10 K this would require a bearing strength of more than 26 kN to prevent deformation.

Conclusion

The primary problem involved with position measurement using rotary encoder and ballscrew is the thermal expansion of the ballscrew. With typical time constants of 1 to 2 hours, thermal expansion causes positioning error in the magnitude of 0.1 mm, depending on the nature of the part program. After every new part program the ballscrew requires approx. 1 hour to attain a thermally stable condition. This also applies for in-

terruptions in machining. A rule of thumb for thermal expansion is that, over the entire length of a cold ballscrew 1 meter in length, the ballscrew grows by approx. 0.5 to 1 µm after every double stroke. This expansion accumulates within the time constant.

As requirements for machine tool accuracy and velocity increase, the role of linear encoders for position measurement grows increasingly important. This should be taken into consideration when deciding on the proper feedback system design.

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Six Degrees of Freedom Error Measuring Device for Linear Motion of Slides

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Abstract

Although laser interferometer measurement system has advantages of measurement range and accuracy, it has some disadvantages when measurement of multi degrees of freedom of motion is required. Because traditional error measurement methods for geometric errors (three straightness and three angular errors) of a slide of machine tools measure error components one at a time. Results of such approaches lead to tedious and time consuming measurements. It may also create an optical path difference and affect the measurement accuracy. In order to identify and compensate for geometric errors of a linearly moving slide in real time processes, an on-line error measurement system for simultaneous detection of six error components is required. Using laser interferometry, beam alignment technique and some optoelectronic components, an on-line measurement system with six degrees of freedom is developed in this paper. Performance verification of the system has been performed on an error generating mechanism. Experimental results show the feasibility of this system for identifying geometric errors of slides.

Keywords : 5 DOF error, 6 DOF error, counter, decimalizing module, error generating mechanism, He-Ne laser, Michelson interferometer, on-line, optoelectronic components, resolution

1. Introduction

As a slide moves linearly in one direction, six degrees of freedom (6 DOF) error motions are generated due to geometric imperfections of guideways, actuator defect, thermal distortion and variation in lubrication conditions, as shown in Fig. 1. The 6 DOF error components will affect the final position of the object. Due to lack of a proper error measuring system, all the calibration methods for geometric errors have relied upon off-line measurement of each of the error components separately.[1] The results of these measurements are not suitable under the real working conditions because of disturbances from load and thermal conditions. In order to calibrate geometric errors of a slide in real time processes,

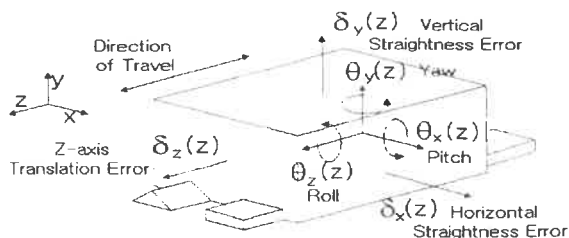


Fig. 1 6 DOF errors of a linear slide

an on-line error measurement system is required. In this paper, a real-time measurement system for simultaneous detection of six error components for a linear slide is studied.

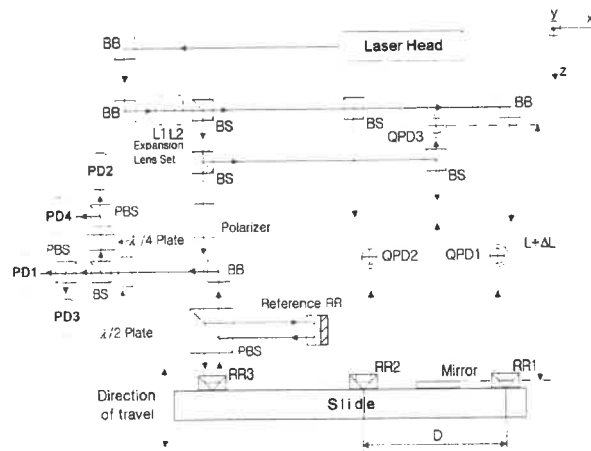


Fig. 2 Schematic diagram of optical system for 6 DOF error measurement

Fig. 2 shows optical parts of the developed 6 DOF measuring system. The basic system is consisted of two parts. One is a modified Michelson interferometer for measurement of linear

displacement and positioning accuracy of a slide. Another is an on-line error measurement system with 5 DOF (two straightness and three angular errors). Measurement principles and experimental results of performance characteristics of the systems are described in this work.

2. Measurement Principles

2.1 Linear Length Measurement

Length measurement part of the 6 DOF measuring system is fabricated through modification of Michelson interferometer as shown in Fig. 3.[2] When the moving retroreflector is translated, reflected beam generates interference signals. Four sinusoidal signals are converted into the $\pi/2$ phase-shifted harmonic signals by retardation plates. Irradiances of interferometric waves detected on photodiodes are given as follows:

$$PD1 : I_1 = I_o + I_c \cos\left(\frac{4\pi}{\lambda} l + \alpha\right) \quad (1)$$

$$PD2 : I_2 = I_o + I_c \cos\left(\frac{4\pi}{\lambda} l + \frac{\pi}{2} + \alpha\right) \quad (2)$$

$$PD3 : I_3 = I_o + I_c \cos\left(\frac{4\pi}{\lambda} l + \pi + \alpha\right) \quad (3)$$

$$PD4 : I_4 = I_o + I_c \cos\left(\frac{4\pi}{\lambda} l + \frac{3\pi}{2} + \alpha\right) \quad (4)$$

where α = initial phase angle
 λ = wavelength of laser beam
 l = moving length
 I_o, I_c = DC and AC components of irradiance

By using differential amplifiers shown in Fig. 4, noise immunized harmonic interference signals A and B are phase-shifted to each other by $\pi/2$ as shown in Fig. 5. Harmonic output signals from the

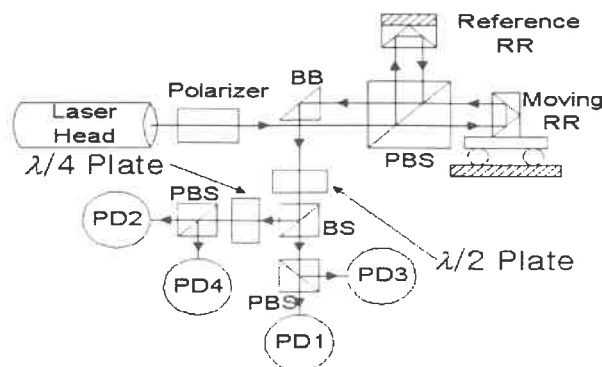


Fig. 3 Optical system for length measurement

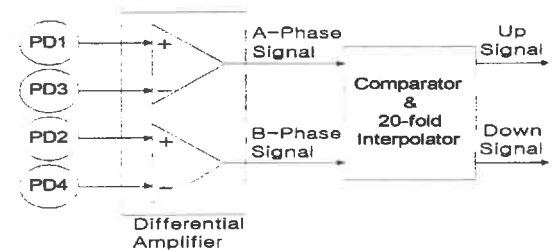
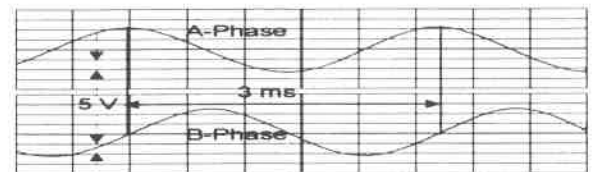
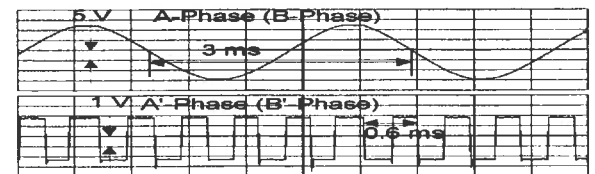


Fig. 4 Interpolation and digitizing signal processor

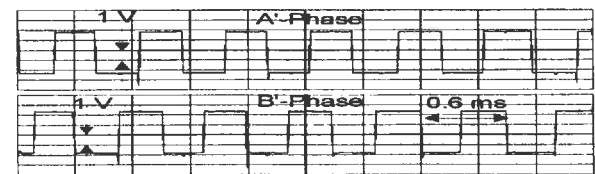
interferometer are processed in comparator and 20-fold interpolator circuits as shown in Fig. 4. In the interpolation and digitizing circuitry the A and B signals are first amplified and then interpolated. Fig. 5(b) shows the result of 5-fold frequency interpolation and digitization as a pulse train. Fig. 5(c) shows square-wave trains of A' and B' with $\pi/2$ phase delay. Fig. 5(d) shows A' pulse and up



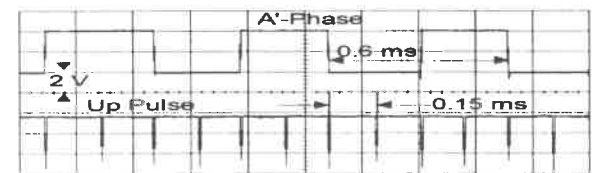
(a) A-Phase and B-Phase Signal



(b) A-Phase and A'-Phase Signal



(c) A'-Phase and B'-Phase Signal



(d) A'-Phase and Up Pulse Signal

Fig. 5 Timing diagram of 20-fold frequency interpolator and digitizing circuit

pulse train generated from edge detectors of A' and B' pulses. In this case a period of up pulse is one twentieth of the half wavelength. Direction discriminative resolution of $\lambda/40$, 16 nm is achieved through the decimal counter with a maximum allowable measurement speed of 600 mm/sec and ranges up to 1.5 meters in the linear

length measurement.

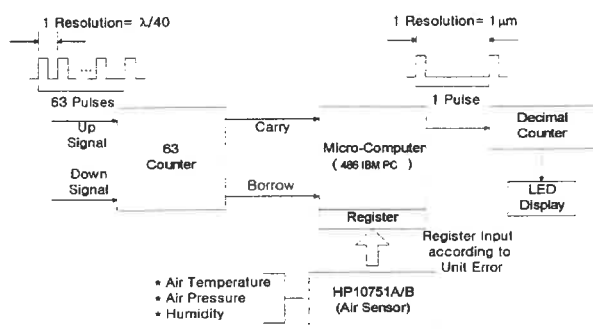


Fig. 6 Schematic diagram of decimalizing module

The absolute accuracy of a laser interferometer in air is determined by the uncertainties in the source frequency and the refractive index of air. A frequency stabilized He-Ne laser is used as a light source. The refractive index of the ambient air is calibrated through the Edlen's formula.[3] By using the cosine error principle and adding an idea of a decimalizing module shown in Fig. 6 to the 16 nm resolution system, resolution of $1 \mu\text{m}$ is also accomplished in length measurements.[4]

2.2 Five Degrees of Freedom Measurement

Using laser alignment technique and some optoelectronic components, an on-line measurement system with 5 DOF is developed as shown in Fig. 7.[1,5] In the system, three reflected beams from the two retroreflectors and the flat mirror are incident on to the quadrant photodiodes (QPDs). Fig. 8 shows arithmetic position sensing unit which calculates spot positions according to linear motion of a slide.[5] Considering kinematic relationships between optics and QPDs, 5 DOF error motions are measured within the linear range of quadrant photodiodes as follows:

$$\text{Horizontal Straightness} = \frac{\Delta x_1 + \Delta x_2}{4} \quad (5)$$

$$\text{Vertical Straightness} = \frac{\Delta y_1 + \Delta y_2}{4} \quad (6)$$

$$\text{Pitch} = \frac{1}{2} \tan^{-1} \left(\frac{\Delta y_3}{L + \Delta L} \right) \quad (7)$$

$$\text{Yaw} = \frac{1}{2} \tan^{-1} \left(\frac{\Delta x_3}{L + \Delta L} \right) \quad (8)$$

$$\text{Roll} = \tan^{-1} \left(\frac{\Delta y_1 - \Delta y_2}{2D} \right) \quad (9)$$

where Δx_i , Δy_i are the spot position of QPDs; L is the initial distance between mirror and QPD3; ΔL is the distance travelled in the moving

direction; D is the distance between RR1 and RR2. Calibration of the arithmetic position sensing unit is needed to determine the coefficients relating the QPD outputs to actual measurement units.

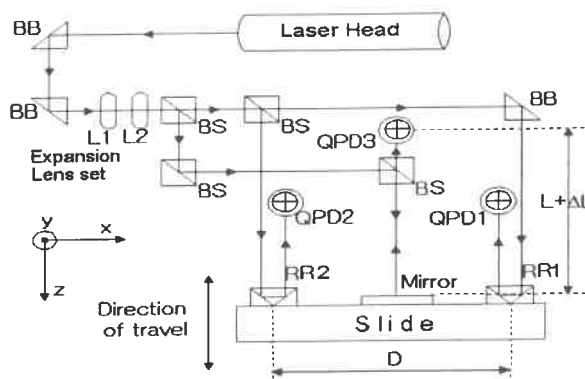


Fig. 7 5 DOF measurement system

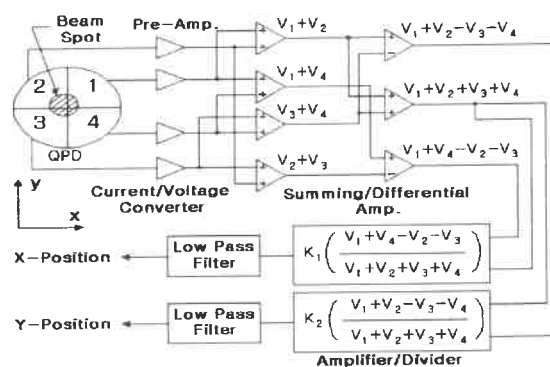


Fig. 8 Block diagram of arithmetic position sensing unit

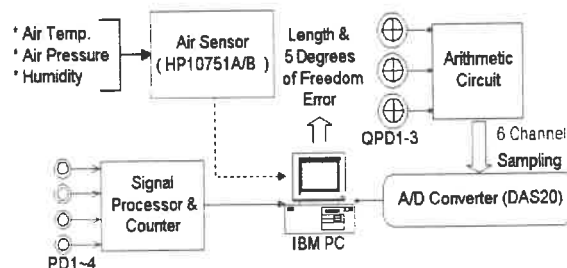


Fig. 9 Signal condition units for 6 DOF measurement

3. Six Degrees of Freedom Error Measurement

Combining the linear length and the 5 DOF measuring modules, 6 DOF measuring device is able to be fabricated as shown in Fig. 2. 6 DOF geometric errors can be simultaneously measured according to the linear position coordinates through the signal conditioner shown in Fig. 9. In order to

evaluate the performance of the system, 5 DOF errors are added to the error generating mechanism shown in Fig. 10 by manipulating screws of A, B and C. A series of experiments have been conducted. Fig. 11 shows the results of them. Within the range of 200 mm, the error of the length measurement is less than $\pm 0.5 \mu\text{m}$ in comparison with the data measured from HP 5529 interferometer at the same time. As the accuracy of the 5 DOF errors created on the error generating mechanism is within the range of $\pm 10 \mu\text{m}$ and $\pm 10 \text{ arcsec}$, linear relationship between the measured outputs and given errors can be confirmed. It verifies the feasibility of the 5 DOF measurement. Fig. 12 shows overall view of the 6 DOF measurement system.

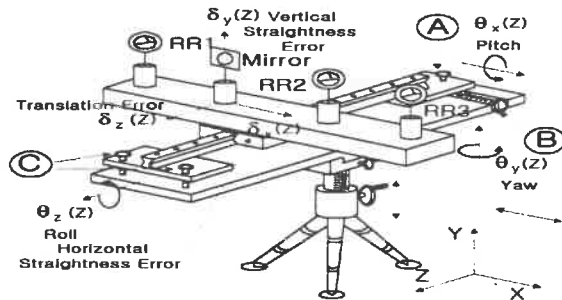


Fig. 10 Error generating mechanism

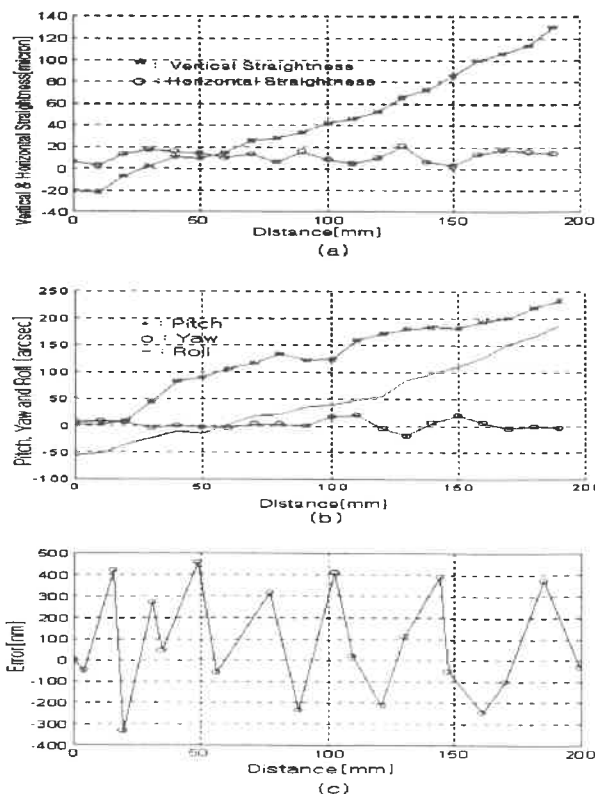


Fig. 11 Geometric errors of the error generating mechanism

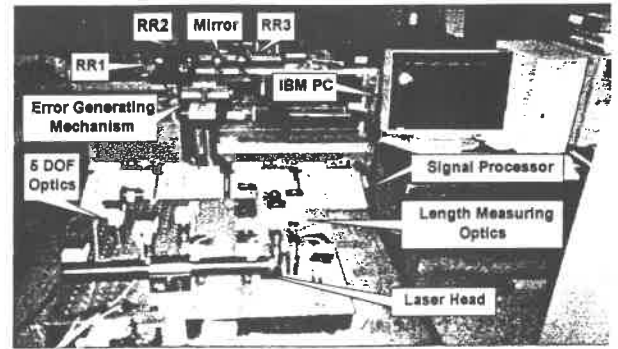


Fig. 12 Overall view of 6 DOF measuring system

4. Conclusions

Using laser interferometry, beam alignment technique and some optoelectronic components, a new on-line 6 DOF error measurement system for linear motion of slides has been developed. Resolution of the length measurement is better than 16 nm and of the straightness is less than $0.5 \mu\text{m}$. Resolution for angular measurement is up to 0.2 arcsec depending on L and D . Performance verification of the system has been performed on the error generating mechanism. Experimental results show the feasibility of this system for identifying geometric errors of slides.

In order to achieve high potential implementation of the developed system, not only must the alignment technique of the 6 DOF system be studied, but also calibration of the system have to be conducted.

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DESIGN AND TESTING OF A CAPACITANCE PROBE CALIBRATOR

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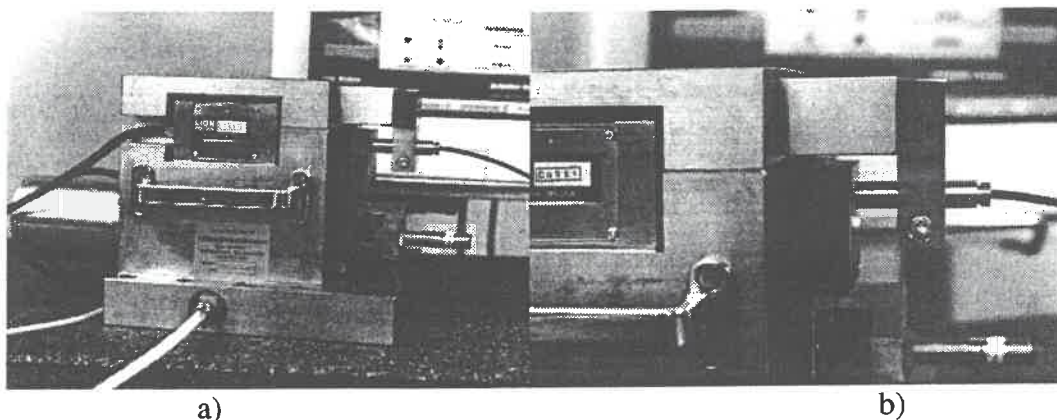


Figure 1. a) *Ultimate Probe Calibrator*; b) *close-up of probe and target*.

Capacitance based displacement sensors are renowned in the precision engineering community for their high resolution and bandwidth. However, calibration of capacitance sensors using available calibrators is very time consuming. Currently available calibration devices typically offer an uncertainty of 130 nm (5 μ in) and require 5 minutes to collect the calibration data. The calibration procedure is also complicated by the need to conduct the calibration at 20°C (68°F), which requires a costly temperature controlled environment. To calibrate capacitance gages, the calibration device must be accurate and fast, yet robust and reasonably insensitive to the environment. This article describes the design of an automated probe calibrator built from INVAR using a precision air slide driven by a linear voice coil with a laser holographic scale which offers uncertainty of 13 nm (0.5 μ in) and a 1 minute cycle time. Issues surrounding the design, testing, and uncertainty of the Ultimate Probe Calibrator are discussed.

Design

The two functional requirements for the calibrator consist of its (i) use on a daily basis and (ii) ability to meet the need of the capacitance sensor customer. Customers of capacitance gages desire high resolution and accuracy without sacrificing cost and availability.

In order to address these issues, several features were considered including the air bearing, drive, scale, probe mounts and targets, and motion control system. The stiff box type bearing, pictured in Figure 1.a), is compensated on the moving slide to minimize sag due to overhung loads. The target cross-section is a 95.3 mm (3.75 in) square, which accommodates large sensing tips and

results in a stiff structure. The normal stiffness of the bearing is 175 MN/m (1,000,000 lb/in) with air consumption of less than 0.6 m³/s (1 scfm) at 0.55 MPa (80 psi). The main structure is fabricated from INVAR to minimize sensitivity to thermal variation and eliminate the need for a temperature-controlled room. The INVAR air slide was designed and built by Professional Instruments Company.

To achieve a non-contact, non-influencing drive, a linear voice coil actuator is used. The actuator is moved when current is passed through the coil, which is placed in a stationary magnetic field. The slide is driven at its center of mass to eliminate “shift” in travel due to unbalanced forces.¹

The voice coil actuator uses a high resolution Sony LASERSCALE as a positioning feedback mechanism. Specifications for the scale are shown in *Table 1*. The LASERSCALE implements a holographic grating system with a pitch of 0.55 μm (21.7 μin) which is located between two glass plates. Moreover, the system is aligned with the probe to reduce Abbé errors.² *Figure 1.b*) shows the probe in line with the slide and the scale. The probe holders are also constructed from INVAR. Finally, the entire system is controlled using a PC-based servo controller. The Windows based calibration software, developed by Lion Precision, Inc., makes the process automated and straightforward.

Table 1. LASERSCALE specifications.

Measuring Length	30 mm
Resolution	0.6895 nm
Repeatability	0.6895 nm 2 σ
Expansion Coefficient	-0.7 x 10 ⁻⁶ /°C
Operating Temperature	0°C to 40°C
Light Source	Semiconductor laser
Detection Principle	Diffraction grating scan

Testing

All uncertainty testing of the calibrator was conducted at the Kansas City Division of Allied Signal Inc. as part of a small business initiative.³ A summary of the uncertainty analysis includes an expanded uncertainty of $U = 12.9 \text{ nm}$ (0.509 μin). The Type A or “random” component of the uncertainty is $u_i = 4.75 \text{ nm}$ (0.187 μin) which is based on 797 degrees of freedom from a data set of 800 points.⁴

This result is obtained from a linear regression model that is fit to the calibrator output to correct its position. *Figure 2.* exhibits the calibrator performance using an interferometer to monitor the position of the calibrator over its entire range. *Equation 1* shows the linear regression model fit to the calibrator according to data taken using the laser interferometer, shown in *Figure 2.b*). In addition, *Figure 2.a*) illustrates a “sawtooth” pattern over a short range of the travel due to the linear interpolation between the 0.55 μm (21.7 μin) marks on the scale.

$$X_c = a + X \cdot (1 + b + c \cdot T) \quad (1)$$

Where X_c is the corrected slide position in μm ,

X is the nominal position according to the scale in μm ,

$a = -0.00196 \mu\text{m}$, $b = 23.93209$, $c = -0.51195 / ^\circ\text{C}$,

and T is the temperature in $^\circ\text{C}$.

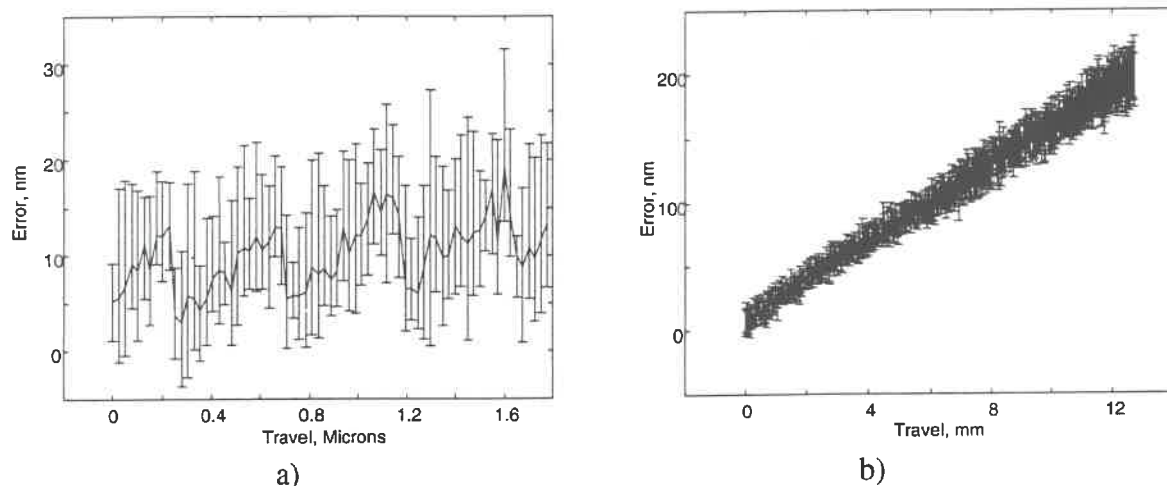


Figure 2. a) Interpolation error; b) total travel error of four tests at 20°C.³

Tests were also conducted in which the interferometer monitored the position of the target over the travel while the calibrator was at three different temperatures, 22.2°C, 24.4°C, and 26.7°C (72°F, 76°F, and 80°F). Figure 3. illustrates the thermal variance of the calibrator for the three tests. The slope of each line is inversely proportional to temperature. Using the scale error and the thermal variance result, an uncertainty of $u_i = 4.75 \text{ nm}$ (0.187 μin) is calculated.

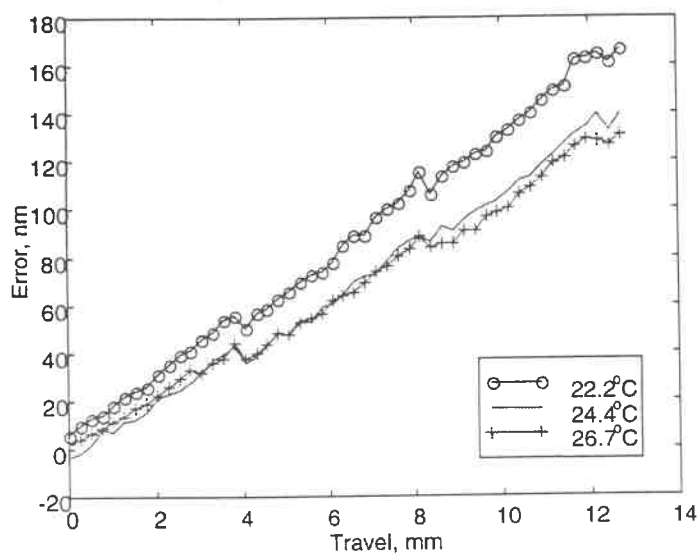


Figure 3. Thermal variance of the calibrator.³

The contributors to “systematic” Type B uncertainty are assumed to be probe temperature, air pressure, relative humidity, laser alignment and air temperature.⁴ Using the uncertainty assigned from each source, the total Type B uncertainty for 12.7 mm (0.50 in) of laser displacement is $u_j = 4.39 \text{ nm}$ (0.173 μin).

Table 3. Type B uncertainty sources.

Source	Source Uncertainty	Base Change per 0.1 ppm	Uncertainty Component (ppm)
Probe Temperature	0.03°C	0.01°C (Steel)	0.300
Air Pressure	0.30 mm Hg	0.28 mm Hg	0.107
Relative Humidity	5%	5%	0.100
Laser Alignment	0.45 mrad	0.45 mrad	0.100
Air Temperature	0.03°C	0.11°C	0.003
Total (u_i)			0.346

The combined standard uncertainty for both Type A and Type B is $u_c = 6.47$ nm (0.254 μ in) using the usual method of “root-sum-of-squares”.⁴ The expanded uncertainty, U , is obtained by multiplying u_c by a coverage factor, k . For normal distributions, $k = 2$ defines an interval with a confidence level of 95 percent.⁴ Therefore, the expanded uncertainty using $k = 2$ is $U = 12.94$ nm (0.509 μ in).

Conclusion

It has been shown in this paper that using a stiff box type air bearing, driven by a non-contacting voice coil using a high resolution scale, can yield a calibrator with minimal uncertainty. The expanded uncertainty was shown to be less than 13 nm, 10 times better than commercially available calibrators. The Ultimate Probe Calibrator has been in use at Lion Precision for two years and the calibration process has been simplified significantly through its use. A second calibrator is currently being built at Professional Instruments Co.

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HIGH-PRECISION DIAMETER MEASUREMENT OF RING AND PLUG GAGES USING A NEW COMPARATOR

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INTRODUCTION

PTB's new comparator for diameter and form has been developed to calibrate the dimension and form of cylinders, spheres and cubes up to 170 mm in length with uncertainties down to $0,01\text{ }\mu\text{m}$, using a contact probing technique [1]. Fig. 1 shows typical calibration standards for industrial metrology with required uncertainties of about $0,1\text{ }\mu\text{m}$, and Fig. 2 shows a ring and a plug gage as parts of a pressure balance [2] with required uncertainties of less than $0,03\text{ }\mu\text{m}$.

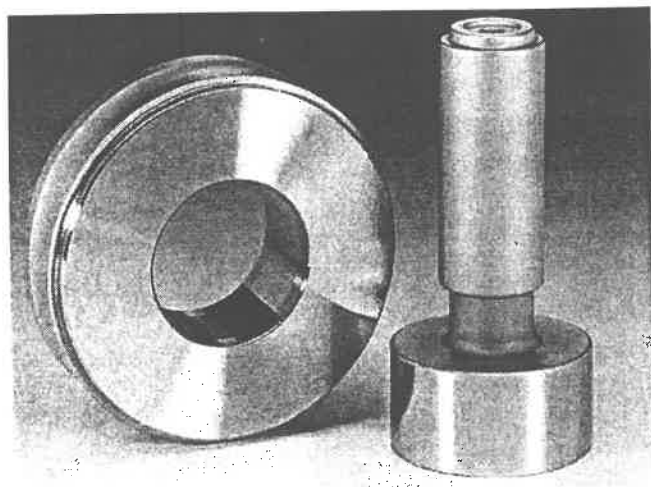


Fig. 1: Ring gage and plug gage as calibration standards for industrial metrology (steel)

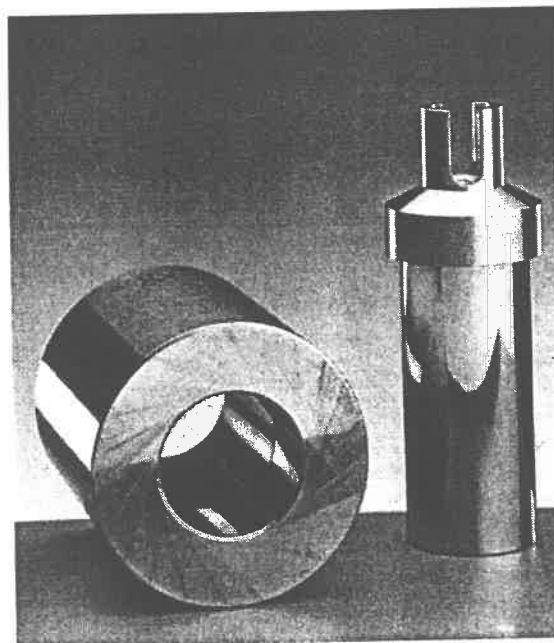


Fig. 2: Ring gage and plug gage of a pressure balance for a research project (tungsten carbide)

To achieve the desired level of uncertainties for objects of high surface quality, the developed measurement device has been equipped with two probing systems. That is why the diameter of the stylus tips (ruby spheres 5 mm in diameter) for outer length measurements need not be known [3], cf. Fig. 3 and next section. Form measurements are carried out by simultaneously probing the object to be measured and a form reference. Disturbing influences due to guiding errors and vibrations are included in both measurement results and can be eliminated [1]. To improve the thermal stability of the measurement device, the measuring frame for the two laser interferometers and the two measuring frames taking up the probing systems are made of invar steel (cf. Fig. 3 and Fig. 4). As a result of the arrangement of the measuring frames, relative motions between object to be measured and length measuring system do not affect the measurements.

The environmental conditions are optimized for high-precision dimensional measurements. The comparator has been installed in the PTB's clean room center to minimize disturbing influences of dust particles on the contact probing technique. The room temperature is almost $20\text{ }^{\circ}\text{C} \pm 50\text{ mK}$ with a variation of the air temperature control by about $\pm 20\text{ mK}$. After strict reduction of the influences of thermal radiation and internal heat sources, the temperature conditions now are excellent. The stability of the material temperature is better than 2 mK per hour and temperature homogeneity over the whole instrument is better than 10 mK . The measurements with the comparator are carried out fully automatically and can be started and observed by the operator via LAN from the control-room or the office in another building.

PRINCIPLE OF OUTER LENGTH MEASUREMENT

A sketch of the new comparator and its principle to measure the outer diameter of a cylinder is shown in Fig. 3. Fig. 4 shows a picture of this part of the comparator.

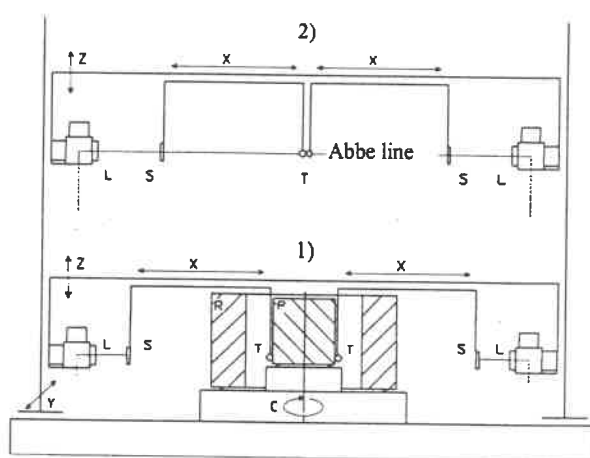


Fig. 3: Measurement of the outer diameter of a cylinder carried out with two probing systems. P cylinder to be measured, R form reference ring, L laser interferometer, S plane mirror, T probing system, X horizontal carriage, Y horizontal frame, Z vertical carriage, C rotating table.

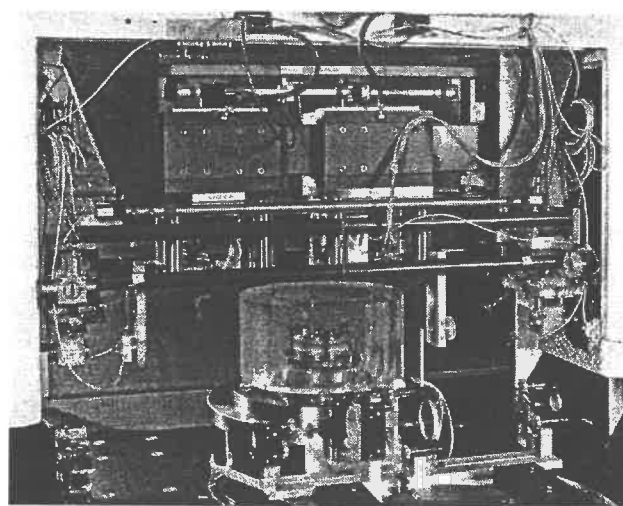


Fig. 4: Measurement guidance of the comparator and rotating table with the form reference ring (ZeroDur) and an object to be measured

The sequence of measurement steps is as follows: First, the stylus tips of the two probing systems (T) touch opposite sides of the cylinder (P). Both carriages (X) are then shifted alternately by steps of one micrometer to about ten micrometer displacement and the sensitivity coefficients of the probing systems (T) are determined from the values furnished by the laser interferometers (L). Secondly, after displacement of the measurement guidance (X) along the z-axis, the two stylus tips touch each other above the cylinder. Only one of the carriages moves for this purpose; so a single sensitivity coefficient for both probing systems is determined from the sum of the values determined by the laser interferometers and the probing systems. The diameter of the cylinder is determined from the four x-positions found for the stylus tips and then corrected to the reference temperature of $20,0\text{ }^{\circ}\text{C}$, taking the coefficient of linear thermal expansion of the cylinder into account.

PRINCIPLE OF INNER LENGTH MEASUREMENT

Inner lengths, too, are measured by means of two probing systems to reduce the size of the measurement frame, thus improving the stability of the measurement assembly compared with measurements with a single probing system. The left probing system contacts the two opposite sides of an object to be measured (P), one after the other. The right probing system contacts an object whose position is fixed in relation to the object to be measured so that displacements between the object to be measured and the length measuring system do not affect the inner diameter measurement.

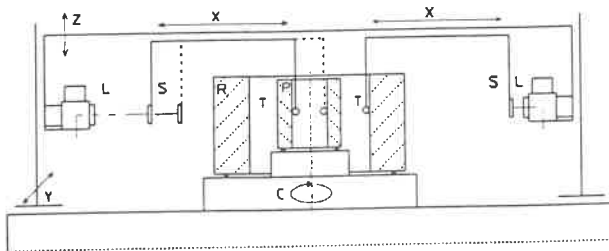


Fig. 5: Measurement of the inner diameter of a ring gage (P) with reference contact at the form reference ring (R)

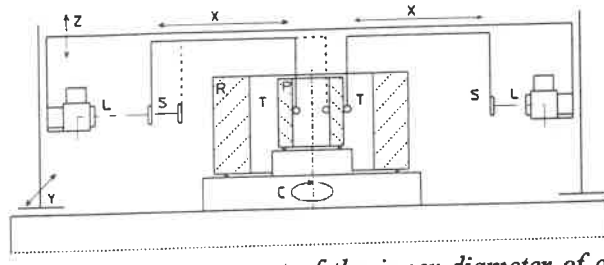


Fig. 6: Measurement of the inner diameter of a ring gage (P) with reference contact at the outside of the ring gage

Fig. 5 shows the measurement of the inner diameter of a ring gage (P) as represented in Fig. 1, that cannot be reproducibly contacted by the right probing system at its outside. The form reference ring (R), therefore, serves as an object with a fixed position. If the ring gage has an outer surface of high quality as shown in Fig. 2, the right probing system will directly contact the object to be measured (Fig. 6). The diameter of the ring gage is determined as described for the outer length measurement, and the stylus tip diameter of the left probing system has to be subtracted.

ADJUSTMENT AND DIAMETER OF CONTACTING SPHERES

For outer length measurements the positions of the zenith points of the left and right stylus tip to each other have to be adjusted in the y- and z-directions within one micrometer (Fig. 7 B). The differences between them are determined by measurements on both sides of a sphere (R) and by subsequent calculation of the centres of the fitted circles (Fig. 7 A).

The diameters of the stylus tips are calibrated using the comparator itself. For this purpose, an additional probing system is set up as an object to be measured in upside-down position (in analogy to sphere (R) in Fig. 7 B). A mirror is fixed to its stylus arm so that the position of this probing system to the y-z plane can be adjusted by the aid of an autocollimator. After the stylus tip diameter of the third probing system has been calibrated, the left probing system (cf. Fig. 5 and Fig. 6) is replaced by the third probing system. This probing system is adjusted to both the y-z plane and the zenith points and can be used now for inner diameter measurements. This direct measurement of the stylus tip diameter reduces the uncertainty of the inner diameter measurements, in contrast to the determination of the stylus tip diameter on a calibrated parallel gage block.

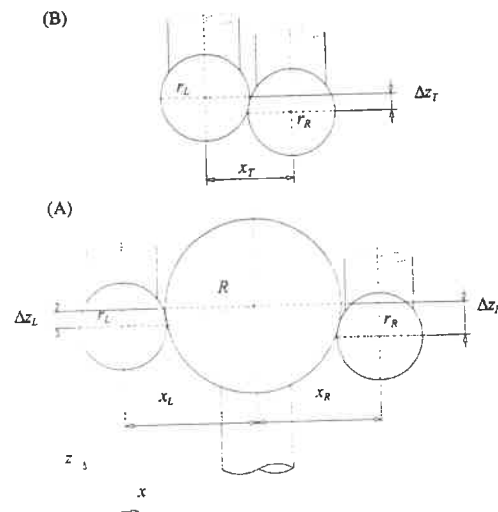


Fig. 7: Measurement of the position of the zenith points of the stylus tips (r_L , r_R) on a sphere (R)

COMPARISON OF GAGE BLOCK MEASUREMENTS

Fig. 8 illustrates the results of length measurements carried out on various parallel gage blocks using the new comparator in comparison with different interference gauge block comparators of PTB with known uncertainty $U = 10$ nm. The values shown are the mean values of several series of measurements, performed over 12 month with the corresponding uncertainty $U (k = 2)$.

The comparison shows good agreement of the obtained measurement results within the range of uncertainty.

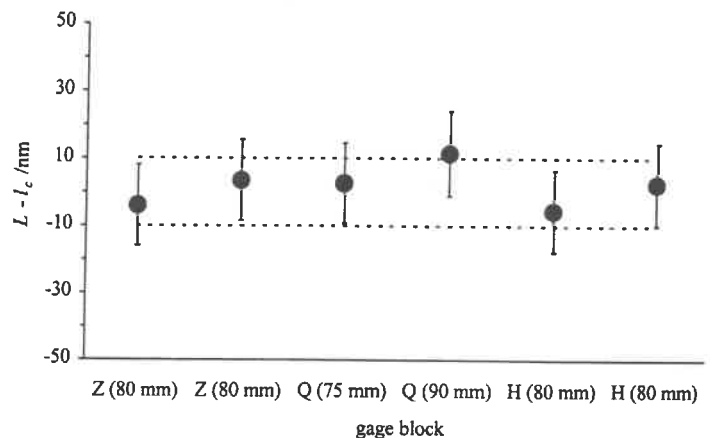


Fig. 8: Differences between the measured and the calibrated length of various parallel gage blocks
Z Zerodur, Q quartz, H tungsten carbide

UNCERTAINTY OF DIAMETER MEASUREMENTS OF RING AND PLUG GAGES

Fig. 9 shows the best measurement capability of diameter measurements with the comparator, expressed in terms of the expanded measurement uncertainty $U (k = 2)$ according to [4]. The differences between the graphs are caused by the length-dependent uncertainty components contributed by the material and air temperature measurements and by some constant uncertainty components. For outer length measurements the stylus tip diameter need not be known. That is why the uncertainty of outer length measurements is clearly lower than the uncertainty of inner length measurements. A detailed uncertainty budget can be found in [3].

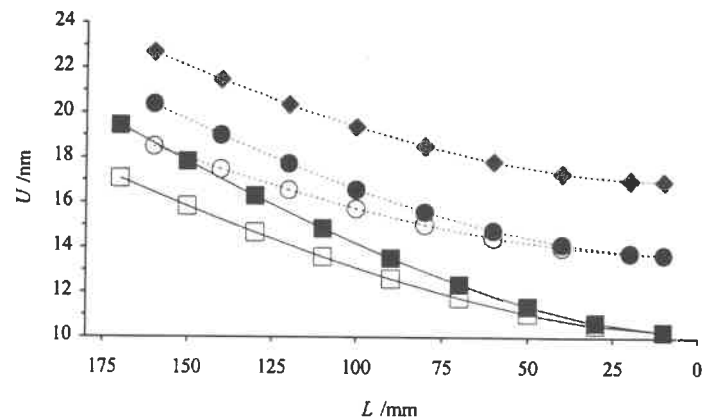


Fig. 9: Uncertainty of length measurements with the new comparator, $\blacksquare \square$ outer length, $\blacklozenge$ inner length with reference contact at the form reference ring, $\bullet \circ$ inner length with reference contact at the outside of the ring gage
 $\square \circ$ tungsten carbide, $\blacksquare \blacklozenge \bullet$ steel

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Design and Control of A Precision Three-axis Stage for AFM Scanner

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1. Introduction

There are recently increasing needs for precision positioning stage in the fields of nanotechnology, especially in AFM(Atomic Force Microscope) [1] and STM(Scanning Tunneling Microscope) [2] for the scanner. No crosscouples and no parasitic motion is essential to investigate topography without any shape distortion. Fast response is essential to investigate topography without any undesirable interaction between the probe tip and the sample surface ; sample scratch. To satisfy these requirements, this paper presents a novel three-axis stage concept as a scanner, it is actuated by PZT and has a monolithic flexible body that is made symmetric as possible to guide the motion of the sample linearly. In this paper the design procedure and its experimental characteristics are presented. The scanning two directions are closed loop controlled using capacitive sensors when scanning range is large(a few microns). In the experiment it is found for the tip-sample distance controller that the control plant is changed by the scanning environment - long range scan and atomic resolution scan. A brief analysis of the control system is presented when the scan range is long and atomic resolution scan.

2. Design

2.1 Modeling of the stage

The stage is shown in figure 1. It is composed of three flexure-mechanism guides which are driven by piezo actuators and are arranged orthogonally. Due to the monolithic and symmetric structure, the stage has no assembly errors and robust to thermal noise. To meet several requirements, optimal design method is used. Therefore dynamic modeling of the stage is necessary and flexure-mechanism guide is modeled using simple force and geometric analysis [3]. The one directional guide mechanism is shown in figure 2. It has eight leaf-flexures, two intermediate body and one main moving body which should be guided linearly. It is proved in our previous work that any parasitic motions such as rotation and linear motion perpendicular to guide direction are not present. The natural frequency of the mechanism can be represented as

$$\therefore \omega_1 = \sqrt{\frac{K}{M}} \quad (1)$$

$$M = \frac{1}{4} M_1 + \frac{1}{4} M_2 + M_3 + 8 \left(\frac{13}{140} \rho b t l + \frac{3}{120} \frac{\rho b t^3}{l} \right)$$

$$K = \frac{2 E b t^3}{l^3}$$

where E is Young's modulus, M_i 's are the masses of the bodies in figure 2, ρ is the mass density and the rests are the dimensions of the leaf-flexure. The analytic model is verified using FE-modeling(IDEAS by SDRC). Triangular mesh is used and the error is within 10%.

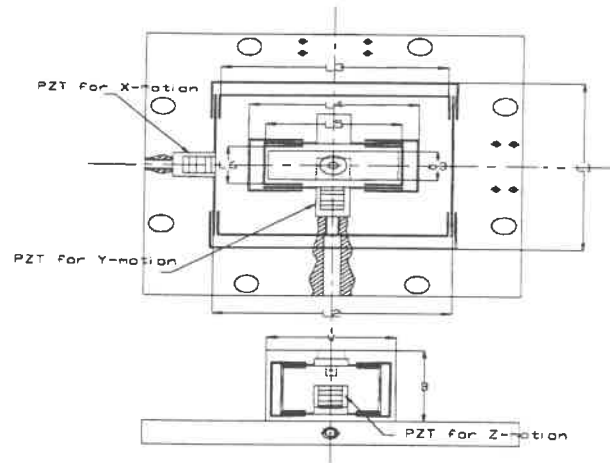


Fig. 1 A schematic of the monolithic precision three-axis stage

2.2 Optimal Design

The design variables are the dimensions of leaf-flexure(t , l , b). The objective of the optimal design is maximizing the first natural frequency of each axis in order to obtain fast reponses. Several constraints are given and the first one is stress limit,

$$g(i) = \frac{3}{2} \frac{E t_i}{l_i^2} q_{i,\max} S_f K_t \leq \sigma_y \quad (i = 1, 2, 3) \quad (2)$$

where S_f : safety factor

$q_{\max}$: maximum displacement

K_t : stress concentration factor

σ_y : yielding stress

i is the index of the stage axis (1 : x direction, 2 : y direction, 3 : z direction)

the second one is limiting the stiffness of the mechanism in order to obtain maximum piezo actuator performance.

$$g(i) = 100 K_j - K_p \leq 0 \quad (i = 4, 5, 6, j = i - 3) \quad (3)$$

where K_i ($i=1,2,3$) is the stiffness of the mechanism in each axis and K_p is the stiffness of the piezo actuator. And there are several dimensional constraints (referring to the figure 1),

$$h(i) = L_i - 70 = 0 \quad (i = 1, 2) \quad (4)$$

$$g(7) = 10t_1 - (L_3 - L_4) \leq 0 \quad (5)$$

$$g(8) = W - L_5 \leq 0 \quad (6)$$

$$g(9) = 3 - (L_6 - b_3) \leq 0 \quad (7)$$

With the above constraints optimal design problem is defined as

$$\text{minimize} \quad -W_z \omega_{1,z}^2 - W_x \omega_{1,x}^2 - W_y \omega_{1,y}^2$$

$$\text{subject to} \quad h(i) = 0 \quad (i = 1, 2)$$

$$g(j) \leq 0 \quad (j = 1, 2, \dots, 9)$$

$$\text{and} \quad 0.5 \leq t_1, t_2, t_3 \leq 3$$

$$5 \leq l_1, l_2 \leq 20, \quad 5 \leq l_3 \leq 10$$

$$10 \leq b \leq 15$$

$$b_3 \geq 12$$

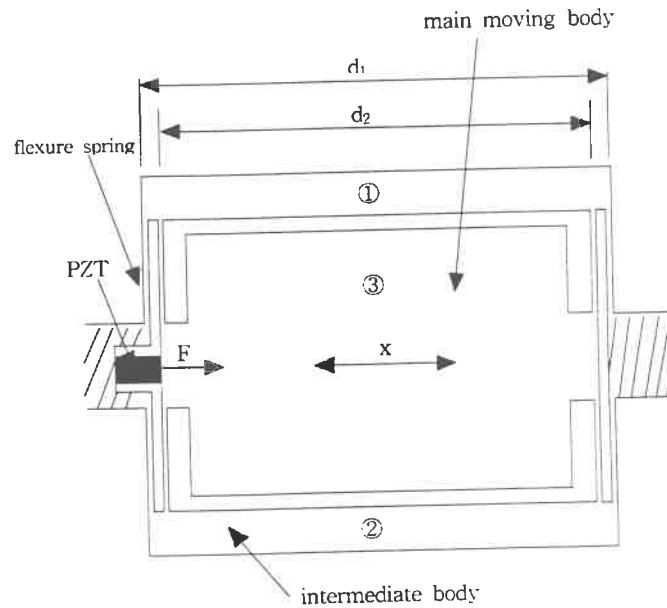


Fig. 2 A schematic of the one axis leaf-flexure guide mechanism

Before manufacturing the stage, the effect due to stage weight is considered. Using simple analysis the effect is proved to be negligible. With the optimal design values, three-axis stage is produced by electrodischarge machining. The stage material is Al 7075-T6 and machining tolerance is within $10\mu\text{m}$.

3. Experimental Results

Scanning two directions are closed loop controlled using capacitive sensors(ADE Corp, $\pm 10 \text{ V} / \mu\text{m}$). The open loop response of the PZT is unsatisfactory to the scanner requirements especially when the scan range is large. The PZT has low damping therefore

the transient response is so vibratory and due to the creep characteristics of the PZT substantial nonlinearity exists. The small cross-couple between the axes enable us to control each axis independently. Because the most dominant plant-element is the flexure mechanism, the controller design is based on the model described above. Mass-spring plant model is used to determine control gains. No overshoot and no vibratory response is observed. Power spectrum of the stage is shown in figure 3. Figure 4 shows multistep closed loop responses in the scanning two directions.

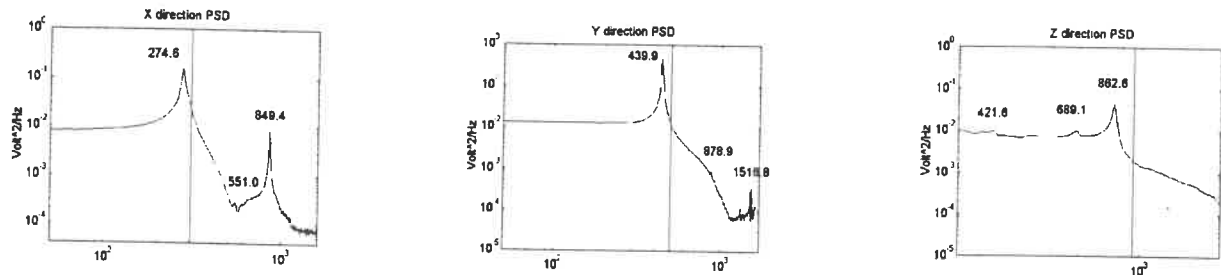


Fig. 3 Power spectral density of the stage (vertical line : solution by analytical method)

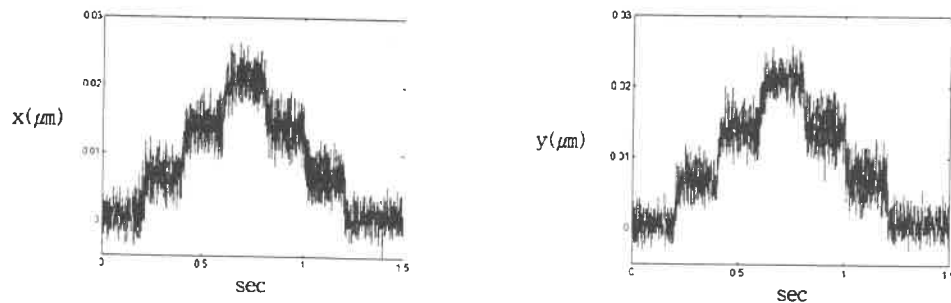


Fig. 4 7nm closed loop step responses (scanning XY-directions)

In our tip-sample distance regulating experiment, it is found that the control plant is changed by the scan environment, especially scan range. During the experiment it is thought that the changes occur when the dominant tip-sample interaction changes. In the atomic scale(nanometer order) scan, the interaction forces, such as van der waals forces, plays a important role in the plant characteristics but when the scan range is large that role importance is decreased. Present research is concentrated on the analysis of this phenomenon. Theoretic and experimental analysis will be done afterwards in the full-paper version. Figure 5 shows the experimental tip-sample distance control results. The experiment is performed attaching another piezo actuator to the z-axis piezo actuator and applying step voltage to the attached piezo actuator while enabling the controller to operate.

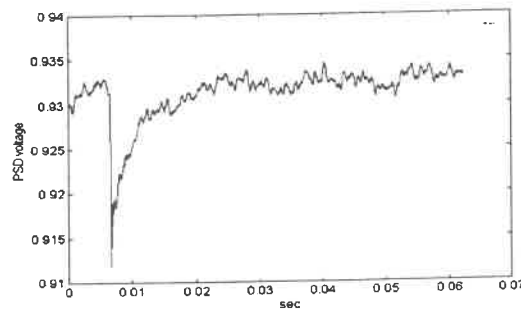


Fig. 5 Experimental tip-sample distance control response

4. Conclusion

To satisfy several requirements of the scanner for AFM, a novel precision three-axis stage is proposed. It has a monolithic flexible body that is made symmetric as possible to guide the motion of the sample linearly. Dynamic modeling is performed and optimal design method is used to maximize first natural frequency of the stage. Based on the model, controller is designed in the scanning directions. Recent research is concentrated on analyzing changes of the plant characteristics due to the scan environment change and this work will present the operator with controller-design guidelines.

Acknowledgement

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Frequency Response of Stylus Probes

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Introduction

Since stylus probing was introduced over sixty years ago, speed and precision of these instruments has been continuously improving. A test bed was designed for the characterization of dynamic performance of stylus based contact probes. This included instruments for both surface finish and form measurement.

This method of testing proved to be useful for not only probe characterization, but also to assess cut-off frequencies of electrical filters within a typical amplitude demodulating system. Based on these tests, the signal conditioning and the mechanical parameters in a stylus system could be assessed and, ultimately, optimized.

To assess the performance, it is necessary to derive an adequate model. Currently, international standards¹ suggest modeling parameters that are separated into flexural stiffness, effective mass of stylus assembly and the damping ratio. In practice the effect of the interaction between the stylus and surface adds considerably to the dynamic performance of many profilers. Indeed, some profilers utilizing free pivots will have a flexural stiffness considerably lower than that of the stylus-specimen interface.

The test bed for experimental evaluation comprises a signal generator, a vibration stage and a spectrum analyzer. Amplitude of vibration is maintained at a constant value using a feedback control loop. The probe under test is mounted on a rigid frame and the stylus is moved to its nominal measuring position on the vibration stage. White noise or swept sine signal is applied to the vibration stage. Data from table and probe under test is compared to generate a frequency response characteristic. Monitoring Coherence between excitation signal to the vibrating stage and probe output provides useful information about the linear regimes of operation for any instrument. Low coherence at near dc (< 1 Hz) may be attributed to the limitations of the test bed. In the frequency region near to resonance, low coherence is commonly a consequence of the sensitivity of the dynamic amplification to small perturbations.

The test bed in its final form is fully automated. A computer with a low cost, data acquisition board is interfaced with the probe. Reports are generated automatically for integration into quality control.

Theory

Contact probes may be approximated by a lumped system as shown in fig 1. Subscripts p and s represent parameters of the probe and specimen respectively. The interface between the stylus tip and specimen surface will provide both stiffness due to Hertzian contact and damping due to the interface energy and internal energy losses². Correspondingly, there will also be stiffness and damping associated with the stylus instrument. In both cases, these are represented by a linear stiffness, k , for elastic forces, b for viscous damping and

h represents losses due to irreversible deformation and interfacial energy dissipation. This latter parameter is associated with hysteresis and, being frequency independent, can be represented mathematically as a complex component of the elastic stiffness. The effective mass can be derived from consideration of the effective kinetic energy of the probe as a function of the tip velocity (i.e. $K.E. = m_e \dot{x}^2 / 2$). From figure 1, the equation governing motion of this system is

$$\begin{aligned} m_e \ddot{x} + (b_p + b_s) \dot{x} - b_s \dot{y} + (k_p + k_s + i(h_p + h_s))x - (k_s + ih_s)y &= 0 \\ b_s \dot{y} - b_s \dot{x} + (k_s + ih_s)y &= F_y \end{aligned} \quad (1)$$

Assuming a closed loop controlled harmonic oscillation of the shaker of the form

$$y = Ae^{i\omega t} \quad (2)$$

where A is amplitude of excitation of the shaker. The complex frequency response, $H(i\omega)$ is given by

$$H(i\omega) = \frac{i \left(2\xi_s \frac{\omega}{\omega_n} + 2 \frac{\chi_s}{\omega_n} \right) + \frac{\omega_s^2}{\omega_n^2}}{\left(1 - \frac{\omega^2}{\omega_n^2} \right) + i \left(2\xi_e \frac{\omega}{\omega_n} + 2 \frac{\chi_e}{\omega_n} \right)} \quad (3)$$

where

$$\begin{aligned} k_e &= k_s + k_p, b_e = b_s + b_p, h_e = h_s + h_p, 2\xi_p \omega_n = \frac{b_p}{m_e}, 2\chi_p \omega_n = \frac{h_p}{m_e} \\ \omega_n^2 &= \frac{k_e}{m_e}, \omega_s^2 = \frac{k_s}{m_e}, 2\xi_e \omega_n = \frac{b_s}{m_e}, 2\xi_s \omega_n = \frac{b_p}{m_e}, 2\chi_s \omega_n = \frac{h_s}{m_e}, 2\chi_e \omega_n = \frac{h_e}{m_e} \end{aligned}$$

Inherent in the above equation is the assumption that the system is linear. Analysis of circular contacts reveal that this is only true for small surface deformations. The magnitude of these can be obtained from the difference in displacements as a function of frequency given by

$$X - Y = A(1 - H(i\omega)) = A \left(\frac{\left(1 - \frac{\omega^2}{\omega_n^2} - \frac{\omega_s^2}{\omega_n^2} \right) + i \left(2\xi_p \frac{\omega}{\omega_n} + 2 \frac{\chi_p}{\omega_n} \right)}{\left(1 - \frac{\omega^2}{\omega_n^2} \right) + i \left(2\xi_e \frac{\omega}{\omega_n} + 2 \frac{\chi_e}{\omega_n} \right)} \right) \quad (4)$$

Space does not permit a detailed discussion of these equations. However, it should be noted that at low frequencies the surface deformation is predominantly hysteretic while the viscous effect increase linearly with frequency. At 'high' frequencies hysteresis can be ignored. It is also apparent that there will be a finite deformation at DC values given by

$$A \sqrt{\frac{\left(1 - \frac{\omega_s^2}{\omega_n^2}\right)^2 + \left(2 \frac{\chi_p}{\omega_n}\right)^2}{1 + \left(2 \frac{\chi_e}{\omega_n}\right)^2}} \quad (5)$$

In the absence of hysteresis losses (permanent deformation) equation (5) represents lost motion due to the finite interface stiffness.

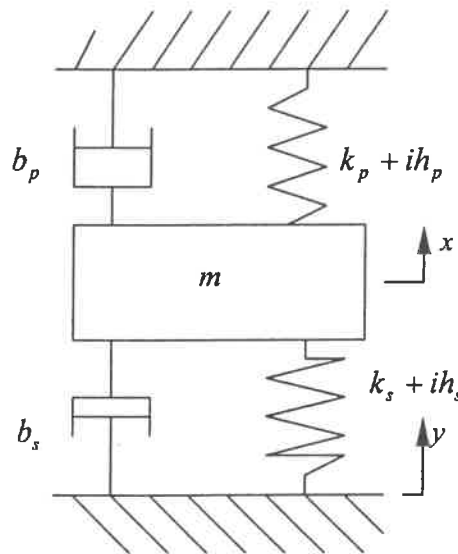


Figure 1: Mathematical model of stylus-probe interaction

Test bed

It is the objective of this brief section to demonstrate the performance and some initial results from the test procedure developed for this study.

The functional requirement of the test bed is to provide a constant amplitude displacement at different frequencies. Typically, commercial vibration stages are specified in terms of acceleration response. At constant amplitude of excitation, the open loop frequency response of shaker table is shown in figure 2. A feedback loop is used to maintain constant output amplitude of the oscillation as input frequencies are varied.

A block diagram representation of the test setup is shown in figure 3. The main components are the vibration table, controller, oscillator, power amplifier and a high bandwidth eddy current, displacement sensor.

During a typical measurement, a constant amplitude displacement swept sine is applied to the moving stage of the vibration table. The frequency response between the displacement of the stage and output of the stylus probe is computed using a digital spectrum analyzer.

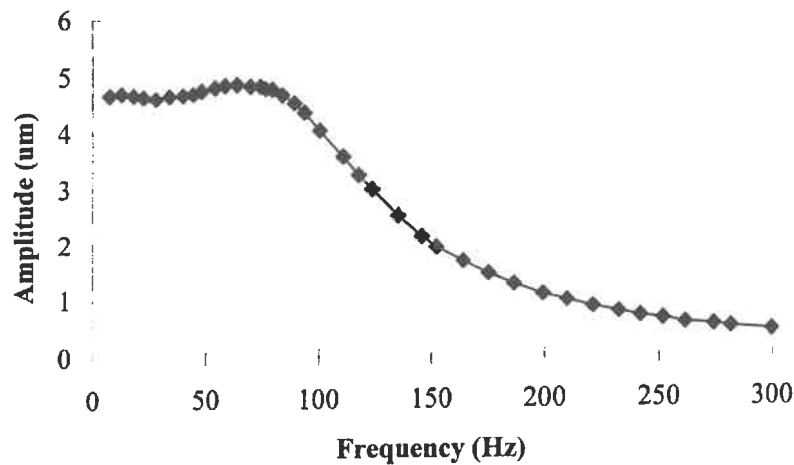


Figure 2: Open loop frequency response of the vibration table

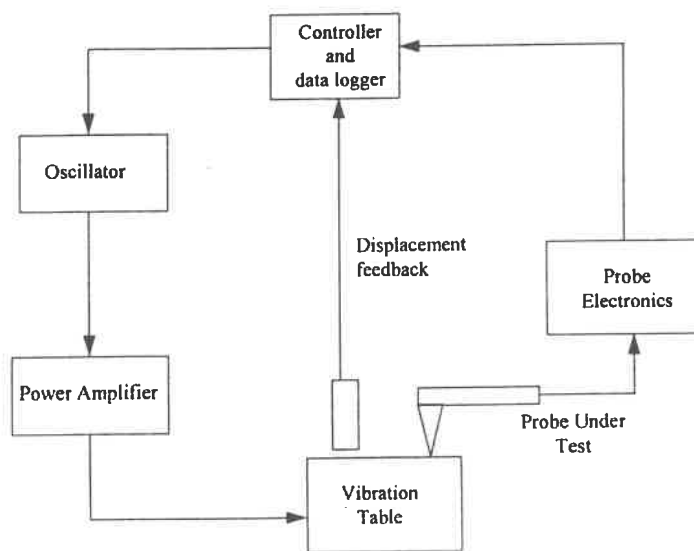


Figure 3: Block diagram of test bed

Results

Three test cases are here presented to illustrate typical measurements and diagnostic capability of this test procedure. For brevity, only magnitude information is presented in this abstract.

Figure 4 shows the frequency response characteristics of three different probe systems. In all experiments the stylus probes have similar geometry are applied to the steel vibration table surface with the same load. Probes 1 and 2 both show responses typical of a single degree of freedom second order system. At frequencies above the first resonance, further peaks are observed but these are not of interest in the current study. Probe 1 has an initial slope caused by the filter cut-off of the demodulator (stylus sensing

is based on frequency demodulation). Probe 2 shows the response of an ideal probe system. This has a 'flat' response over a relatively broad bandwidth, in this case remaining constant to within a few percent up to a frequency of 300 Hz.

Probe 3 illustrates a defective probe. However, in this case, the low frequency response is constant at lower frequencies and would produce acceptable measurements. However, at higher frequencies the response characteristic becomes erratic and would certainly produce erroneous measurements if signals above 200 Hz were to be used.

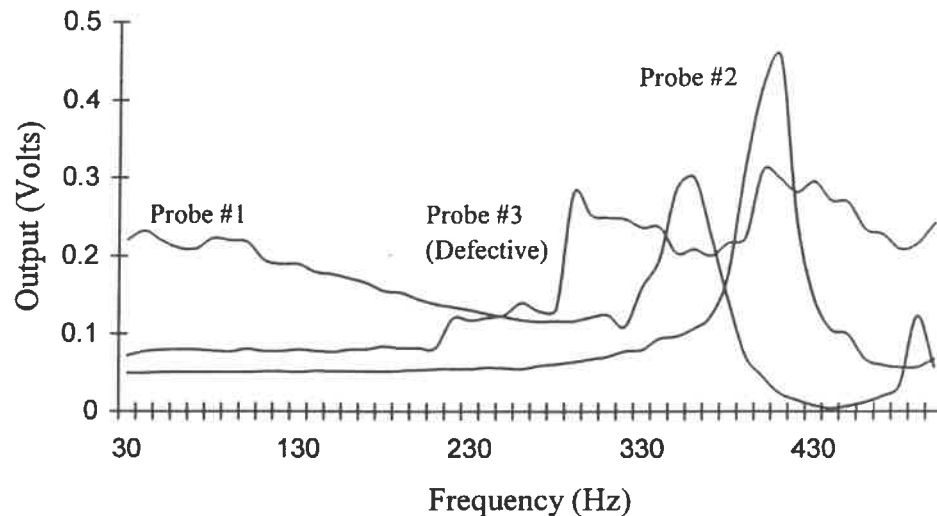


Figure 4: Frequency response of a stylus probe

Conclusions

This brief abstract presents the design of a test bed for the frequency-based assessment of stylus based systems. It has been shown that probe systems operate over finite frequency ranges and that these may vary from probe to probe even with nominally similar systems. Some future issues to be addressed using this system are;

1. Diagnostic value of the coherence function for the identification of 'defective' probes
2. The effect of parameters such as probe tip and specimen geometry, nominal applied load and specimen material.
3. Effects of contact on the probes 'free' natural frequency
4. Characterization techniques for the assessment of large amplitude response of stylus probes

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Development of a Metrology Instrument for Mapping the Crystallographic Axis in Large Optics*

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Abstract

A metrology instrument has been developed to scan crystals and map the peak tuning angles for frequency conversion from the infrared to the ultra violet over large apertures. The need for such a device emerged from the National Ignition Facility (NIF) program where frequency conversion crystals have been found to have significant crystallographic axis wander at the large NIF aperture size of 41 cm square. With only limited access to a large aperture laser system capable of testing these crystals, scientists have been unable to determine which crystal life-cycle components most affect these angular anomalies. A system that can scan crystals with a small diameter probe laser beam and deliver microradian accuracy and repeatability from probe point to probe point is needed. The Crystal Alignment Verification Equipment (CAVE) is the instrument designed to meet these needs and fit into the budget and time constraints of the ongoing NIF development.

In order to measure NIF crystals, the CAVE has a workspace of 50 x 50 cm and an angular measurement accuracy of 10 μ rad. Other precision requirements are probe beam energy measurement to 2% of peak, thermal control to $20 \pm 0.1^\circ\text{C}$ around the crystal, crystal mounting surface flatness of 1 μm over 40 cm square, and clean operations to Class 100 standards. Crystals are measured in a vertical position in a kinematic mount capable of tuning the crystal to 1 μ rad. The mirrors steering the probe beam can be aligned to the same precision.

Making tip/tilt mounts with microradian level adjustment is relatively commonplace. The real precision engineering challenge of the CAVE system is maintaining the angular alignment accuracy of the probe laser relative to the crystal for each spatial position to be measured. The design team determined that a precision XY stage with the required workspace and angular accuracy would be prohibitive to develop under the given tight time constraints. Instead the CAVE uses commercially available slides and makes up for their inaccuracies with metrology. The key to the CAVE device is referencing all angular measurements relative to a master reference surface. The angles of the probe laser and the crystal mount are measured relative to a 61 cm optical flat for every spatial location. Autocollimation and imaging are used to achieve the microradian angular precision. Slide errors are removed by alignment of the crystal and the beam for each new probe point through active feedback from the autocollimation system.

The first prototype CAVE system is being constructed and tested at Lawrence Livermore National Laboratory. The mechanical and optical errors are being measured and analyzed to verify the critical performance specification of 10 μ rad. Others facing the same challenge of designing point-to-point probe system for measuring optical axes can use the knowledge gained from this design.

Keywords: crystallographic axis, autocollimation, metrology frame, optical mounting, optical testing, and National Ignition Facility

1. Introduction

In order to meet current NIF specifications, third harmonic conversion efficiencies of greater than 80% of the input pulse peak irradiance are required. The NIF converter design sends an input irradiance of 3.3 GW/cm² through a pair of 410 mm square crystals, a type I 11 mm thick KDP second harmonic generator (SHG) and a type II 9.2 mm thick KD*P third harmonic generator (THG)¹. The ideal conversion efficiency for this system including crystal absorption is 90.8%¹. This sets a total conversion error budget of 9.35% of which angular sensitivity is a large contributor, Table 1. Figure 1, shows the microradian level tuning that must be maintained. The solid curve represents

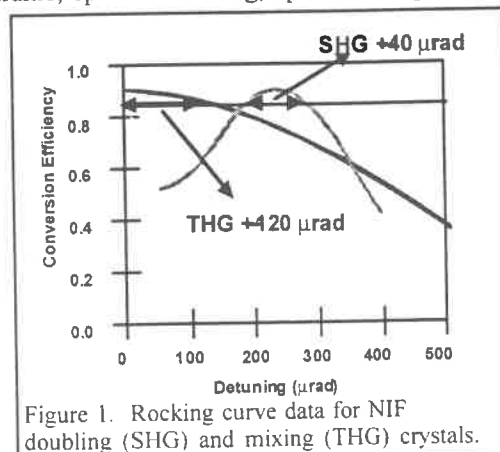


Figure 1. Rocking curve data for NIF doubling (SHG) and mixing (THG) crystals.

the detuning of the SHG to find the optimal mixture of 1ω and 2ω light for maximum 3ω conversion through the THG which gives a $40\text{ }\mu\text{radian}$ error band for 85% conversion. The dotted line shows the detuning of the THG given the optimum mixture from the SHG (120 mradian error band for 85% conversion). Evidence from current experiments at Lawrence Livermore National Laboratory indicate that these curves, known as rocking curves, can shift for different spatial locations on a crystal. Significant spatial non-uniformity in frequency conversion has been observed with the 37 cm aperture Beamlet experiment.ⁱⁱⁱ A device is needed to measure these conversion variations directly by measuring multiple rocking curves over an entire crystal.

Table 1. NIF frequency converter error budget.ⁱ

Source of Efficiency Reduction	Magnitude
1 ω Beam Losses	2.2 %
Coating loss	4.26
Loss terms	0.76 %
Angular Sensitivity Terms	1.44%
CAVE offsets	0.69 %

The CAVE device has been designed and is being constructed to fill the role of measuring harmonic conversion over entire crystal apertures. The final version of CAVE will be used for crystal alignment verification of the NIF Final Optics Assemblies before they are installed to the laser system.^{iv}

2. Design

The CAVE device is designed to measure sample crystals up to the 41 cm square NIF size. A cell is made to hold both a doubling and tripling crystal or a single crystal, Figure 2a. In order to minimize distortion of the crystal from mounting, both mounting surfaces are diamond flycut to 1 micron flat. The cell flanges give an even pre-load of 2 N/cm to hold the crystal edges against the mounting surface. Crystals are held in a vertical position to eliminate gravity sag as they are scanned. Three actuators kinematically attach to the cell tip, and tilt the crystal and are capable of tuning the angles to less than $0.5\text{ }\mu\text{radians}$.

All angle measurements are referenced to a 610 mm optical flat that is mounted in line with the crystals, Figure 2b. This reference flat is 2.5 waves, at 632 nm, and has angular variations calibrated to $0.8\text{ }\mu\text{radians}$ with a look up table. The flat serves as the null and the angles of the laser and the crystal mount are measured relative to its normal.

The laser autocollimator, Figure 2e and Figure 3b, contains a CCD camera that can detect the relative angle between the probe laser beam and the reference flat's surface normal. The autocollimator camera records the focal spot's position from the probe laser beam and the reference laser beam's reflection off the reference flat.

The diamond turned surface the crystals are mounted against extends below the crystal to provide a reference surface for measuring the angle of the crystal mount. The crystal autocollimators, Figure 3a,

measure the angle of the diamond turned surface relative to the 610 mm reference flat. The autocollimators remove the stage errors as the crystal cell moves horizontally.

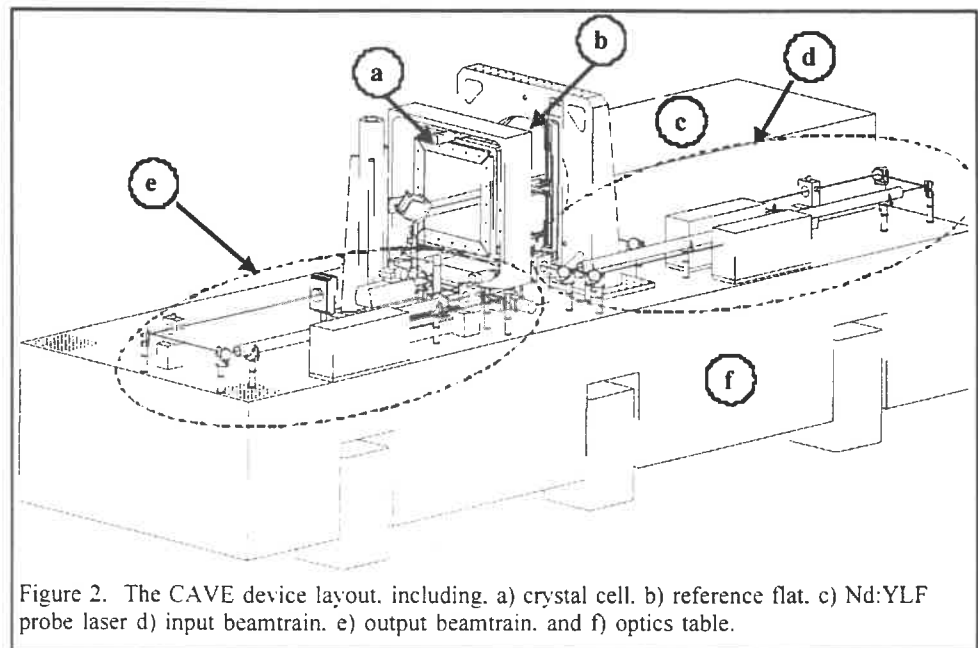


Figure 2. The CAVE device layout, including. a) crystal cell. b) reference flat. c) Nd:YLF probe laser d) input beamtrain. e) output beamtrain. and f) optics table.

The Nd:YLF probe laser, Figure 2c, delivers a fluence of up to 4 GW/cm^2 at the crystal at 1053 nm wavelength. The laser has a pulse length of 100 picoseconds, a beam diameter of 5 mm at the crystal test surface, and a maximum repetition rate of 10 Hz. With this short pulse length, the crystal can be probed with many high fluence pulses without significant thermal changes ($\pm 0.1^\circ\text{C}$). The input optics, Figure 2d, relay image the laser beam from the last amplifier rod to the crystal surface. A portion of the beam is split off to the reference crystals and reference energy meters. These reference crystals converted energy measurements is used to normalize the output energy from the sample crystal due to variations in the pulse shape, energy or beam profile.

To scan the crystal in the vertical axis, periscopes translate the beam vertically on the input and output sides of the crystal. Two optical trombones maintain the relay path length for each vertical movement. The laser autocollimator monitors the alignment of the output laser beam, Figure 2e. Tip and tilt adjustment of the bottom periscope mirror on the input side controls the pointing of the laser. On the output side another tip and tilt adjustment of the bottom mirror of the output periscope aligns the autocollimator with the reference flat. Further down the output beam the output energy meter measures the pulse conversion. Additional crystal diagnostics have been added to the output side to image the beam at the crystal surface to observe local variations in conversion efficiency.

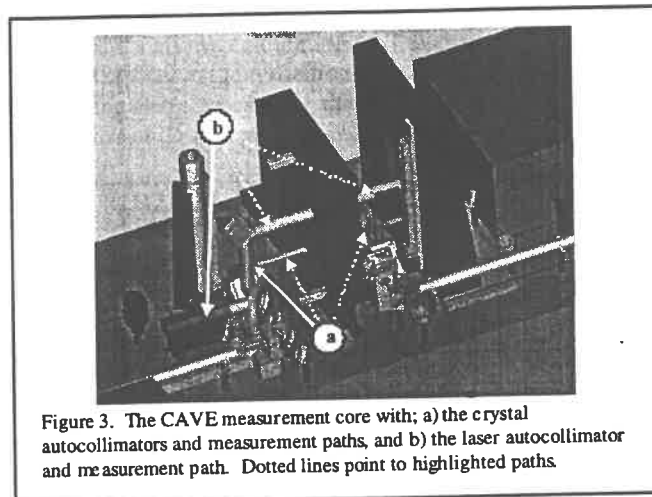


Figure 3. The CAVE measurement core with; a) the crystal autocollimators and measurement paths, and b) the laser autocollimator and measurement path. Dotted lines point to highlighted paths.

Several environmental controls need to be maintained for the CAVE device to operate effectively. The optics table, Figure 2f, is on air isolators to attenuate vibrations above 2 Hz. The table is housed in a Class 100 clean room enclosure that maintains temperature of $20 \pm 0.1^\circ\text{C}$.

The operating modes of the CAVE device are energy meter calibration, crystal scanning and rocking curve measurement. An absolute energy meter is inserted at several places in the beamtrain to calibrate the input, output and reference energy meters to take out system losses. Once crystals are loaded in the cell, the main stage moves to the first horizontal scanning position and the cell is angularly aligned. For vertical scanning the beam is moved with the periscopes and aligned relative to the reference flat at the new position. When a position is set, the cell is tipped or tilted to the desired angle, a laser pulse is released and all data is recorded. This is repeated for all of the points on the rocking curve. The final output of each experiment is an array of horizontal location, vertical location and probe angle with the corresponding normalized conversion energy output.

3. Instrument Error Budget

The final machine must measure rocking curves at multiple locations on a KDP crystal with a high level of repeatability. The error bars for a single shot (single point on a rocking curve) have been designed to be less than 10 $\mu\text{radians}$ in angle and 2% of total energy. Many of the large angular errors inherent in machine motion are taken out with the metrology control, most notably with the autocollimators which monitor the angles of the crystal and the laser relative to the reference flat. Table 2. shows the angular errors and the root mean square total.

Table 2. CAVE Angle Measurement Error Budget .

Error Component	Net Error μrad	RSS tot
Reference Flat	1.0	
Clipping and field effects	1.1	
Laser Autocollimator centroid	5	
Resolution and Stability	1.7	
Net Beam relative to Ref. flat		5.48
Mount flatness	1.0	
Crystal Autocol. centroid	5	
Resolution and stability	2.3	
Mount relative to Ref. flat		5.59
Laser beam pointing		5
Scanning errors		1.6
Tot. RSS system errors		9.42

4. Instrument Component and System Verifications

Table 2 gives a summary of the angular error budget for CAVE. Each of these error components needs to be validated to ensure the system error budget is met. This process is under way now and is a time consuming process to verify component performance at micro-radian accuracies.

The reference flat and FOC mount surface figures have been measured with interferometry. The reference flat's figure was measured to allow correction for errors caused by using different portions of its surface as reference surfaces. The laser autocollimator looks at different heights on the reference flat as the laser beam is moved to different heights in the crystal. The flat mounting surface for the crystal was measured to determine its deviation from a flat surface. This allows calculation of the errors introduced by clamping crystals. It also allows correction to errors the crystal angle measurement caused by the autocollimators viewing different portions of the mounting flat's area below the crystal, Figure 3a. This data combined with look up tables results in errors for these components under 1 μ radian.

As an example of the measurement capability of the system now, Figure 4 shows a single point rocking curve. This shows the primary and secondary peaks in the rocking curve and it is normalized to the 2w reference input power, and each point was averaged from three points. The rocking curve does show the expected hyperbolic sine squared function shape that is typical of rocking curves at medium to low power, under 1 GW/cm². Work is continuing on the verification of the system error budget.

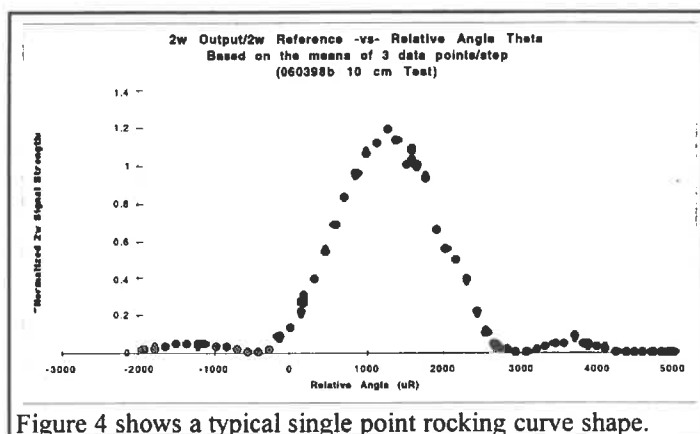


Figure 4 shows a typical single point rocking curve shape.

5. Summary

The first CAVE prototype is currently being constructed and tested at Lawrence Livermore National Laboratory. The device scans KDP crystals and measures conversion efficiency vs. rocking angle for any spatial location. The CAVE is designed for accuracy of 10 μ rad in phase matching angle measurement and 2% in conversion energy measurement. The machine errors will be tested and analyzed, leading to a final design that will be used for qualification of the NIF frequency converters. The immediate use for the CAVE prototype will be to help advance crystal development to reach the NIF conversion goals.

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Vector Touch Sensor Probe

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Introduction

Current contact sensors produce a single pulse or proportionate signal upon being pushed against a surface. The objective of the current sensor development is to determine the vector normal when a sphere is contacted with a solid surface. It is envisaged that such a probe will be used in co-ordinate measuring machines and other applications requiring contact based information. Present touch trigger probes that produce a single co-ordinate set per measurement require two points of contact to obtain an estimate of local slope. In contrast the vector touch sensor probe will require only a single point of measurement, thereby doubling the Nyquist frequency. Subsequently, for scanning applications, there will be a potential doubling of the sampling speed for a given accuracy of profile reconstruction. Characterisation of the current sensor utilises both phase lock loop techniques and piezo-electric ceramic (PZT) sensor/actuator pairs. This abstract presents results from the characterisation of a probe in two-dimensional space and subsequent design of a second-generation probe for improved sensitivity and ease of use.

Principle of operation

Experimental apparatus for the 3 dimensional characterisation of a resonating touch based sensor was developed. A first generation probe shown in figure 1 consists of a rectangular rod, two PZT actuator-sensor pairs and a spherical stylus.

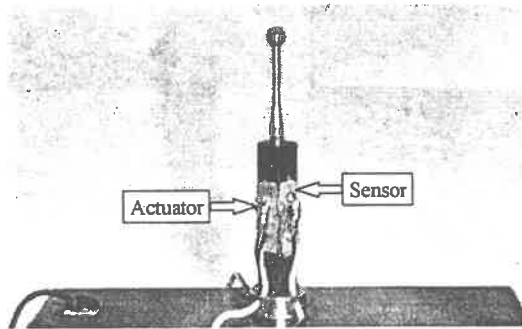


Figure 1: First Generation Probe

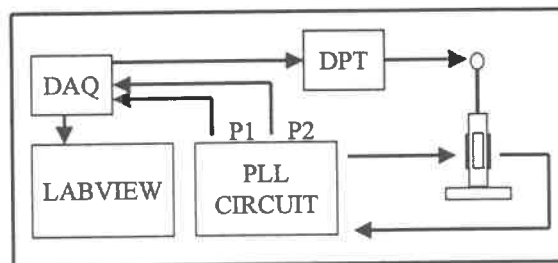


Figure 2: Experimental Procedure

Sweeping the input excitation across any resonance frequency of the probe results in an amplification of the output signals from the sensors while the phase-lag changes by 180 degrees. These points in the frequency response are of critical interest for a touch-based sensor. Effectively, the rate at which phase changes when sweeping across a resonance determines the sensitivity of the sensor. The magnitude of the phase change when the probe is contacted with a surface is dependent upon the mechanical properties of the contact and the applied force, Vidic *et al.*, 1998. Also, more importantly for the purpose of this study, the phase shift is dependent upon the direction of the externally applied force with respect to the line of action between the actuator input and sensor output. For example, the sensitivity of a system's phase is the greatest when an externally applied force is co-linear to the line between a given actuator and sensor pair. This

physical phenomenon provides a potential means for the design of a vectored touch probe.

To evaluate the vector response characteristic of such a probe a prototype sensor consisting of a 7x5x31mm stainless steel cantilever threaded to a brass interface was produced, see figure 1. The steel to brass interface at the probe mount results in a sudden change in velocity of wave propagation thereby enhancing acoustic reflectance. A coordinate measuring machine probe tip with a 4 mm sapphire sphere is mounted using an M2 thread into the opposite end of the cantilever. Two pairs of piezoelectric actuators (EC64) and sensors (EC65) are cemented to the cantilever. Each PZT pair is oriented to produce orthogonal lateral excitation and sensing. To separate the responses in different contact orientation, it is imperative that both signals occur at different frequencies. In this case, the two frequencies were offset by 4 kHz to produce a first mode frequency of 31 kHz and 35 kHz, respectively. Utilising Phase lock Loop (PLL) circuitry, the phase or frequency deviation may be monitored. Observing phase shift as frequency is held constant and keeping phase constant as frequency is shifted correspond to two different modes of operation for detecting contact of the probe sphere. These two locking techniques are referred to as Mode 1 and Mode 2 respectively. Figure 2 shows a block diagram of the system for monitoring mode 1 or 2 signals from each sensor-actuator pair. Contact between a specimen surface and the probe tip is achieved using Labview™ software to incrementally position a displacement positioning transducer (a Queensgate Instruments DPT). During current experiments, each axis, labelled P1 and P2 in subsequent figures, on the probe is excited at a constant frequency and phase is monitored. During experiments phase data is collected as the DPT incrementally displaces the specimen surface towards the probe. It is also possible to lock into a selected phase and monitor the subsequent frequency shift during contact. Upon contacting and retracting the specimen, the probe is rotated about the vertical axis and the procedure repeated. Thus, a two dimensional polar plot may be achieved to show phase sensitivity as a function of the angle of contact about the spherical probe.

Current research aims to improve the sensitivity and stability of the first generation probes. As well as the interfaces between the probe and sensor body, the first generation probes' were rigidly mounted by threading the steel body onto a brass interface. This design lowers the sensitivity of the probe because the mounting is not at a vibration node. Theoretically a free rectangular bar has multiple mode shapes along the length and, for each mode, there are corresponding node locations, figure 3.

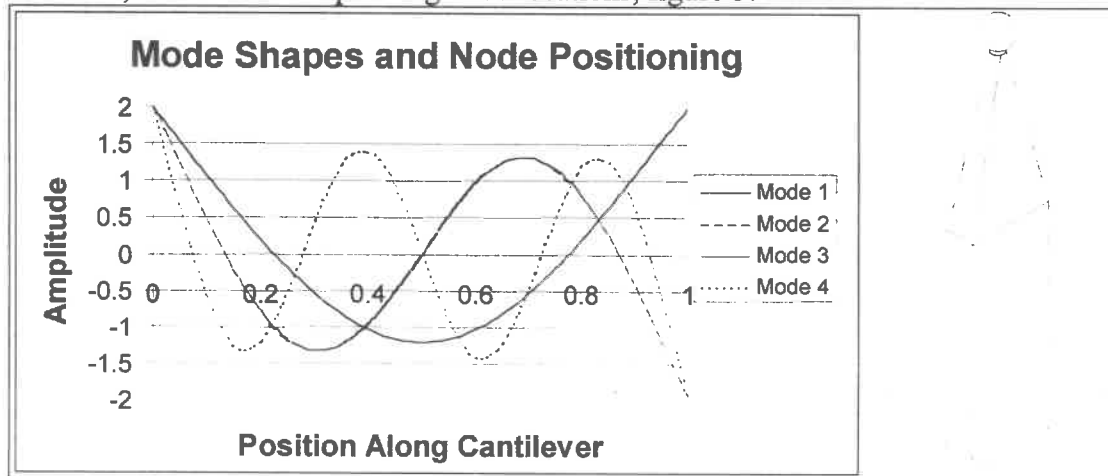


Figure 3: Rectangular bar modal shapes and second generation probe

The node locations for a given mode occur at the intersection of the x-axis. It is at these nodal locations that an optimal mounting should be placed. Also shown in figure 3

is the design for a second-generation probe. This is constructed from a single monolith and consists of a beam of rectangular cross-section with a shoulder at a nodal location at one end and a taper at the other. At the end of the tapered portion there is a small rod onto which the ruby probe head is attached using two-part epoxy. The actuator-sensor pairs are attached to the rectangular bar at the bottom (in this figure) and, in the process of mounting, electrical contacts are automatically made. Apart from the beryllium copper spring electrical contacts the only other interface is that of the shoulder clamp positioned at a nodal point.

Experimental Evaluation

A differential position transducer (DPT) with a range of 15 μm and resolution of 1 nm was used to characterise externally applied force and signal phase offset. A two axis (x - y) stage provided initial positioning of the DPT and probe's sphere and, once positioned, was rigidly clamped. Frequencies of interest were identified using a capacitance based force sensor and fixture (not shown) to determine frequency response as function of applied force. Simultaneously measuring displacement of the DPT and phase changes (using a Stanford SR650 Lock-In-Amplifier), it was possible to determine phase shift as a function of applied contact force. Figures 4, 5 and 6 demonstrate the phase change in both sensor axes as the DPT applies the specimen surface (in this case a steel gage block) at three orientations on the probe sphere. Note that A, S, P1 and P2 in the figures correspond to the Actuator, Sensor, Position 1 and Position 2, respectively.

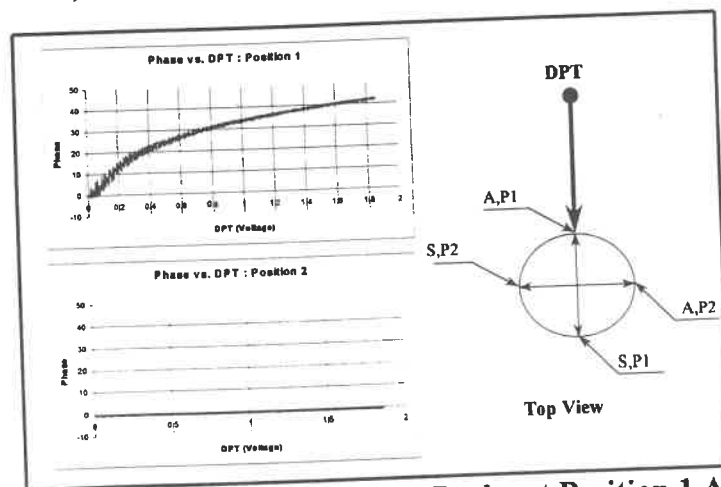


Figure 4: DPT Applied to Spherical Probe at Position 1 Actuator

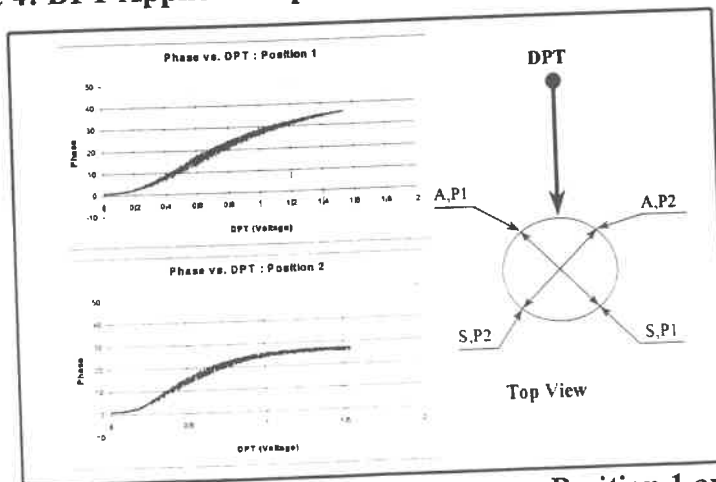


Figure 5: DPT applied to spherical probe between Position 1 and 2 Actuator

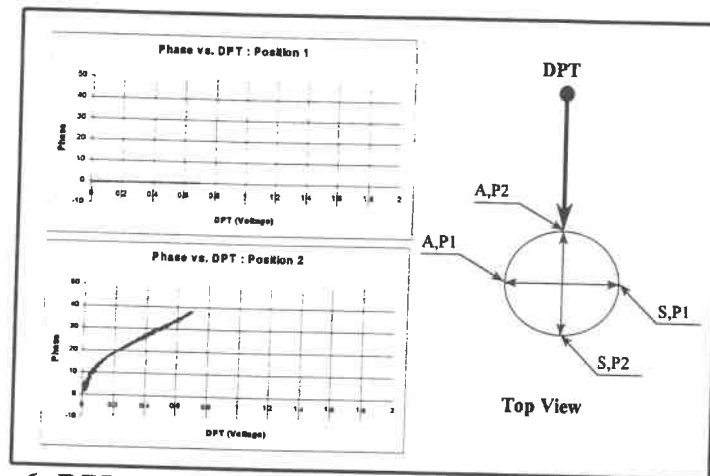


Figure 6: DPT applied to spherical probe at Position 2 Actuator

From the above figures, it is clear that there is a good correlation between phase shift as a function of applied load and direction of the contact normal.

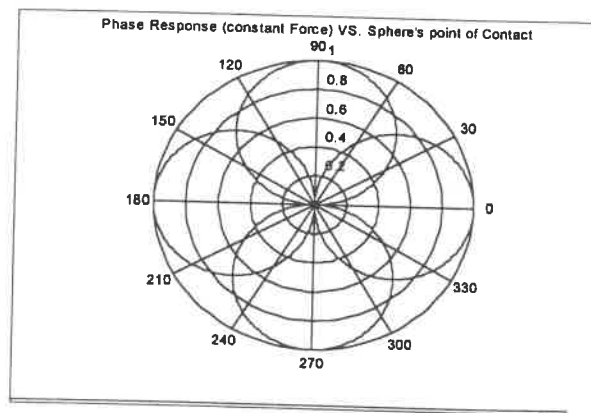


Figure 7: Ideal Phase Sensitivity

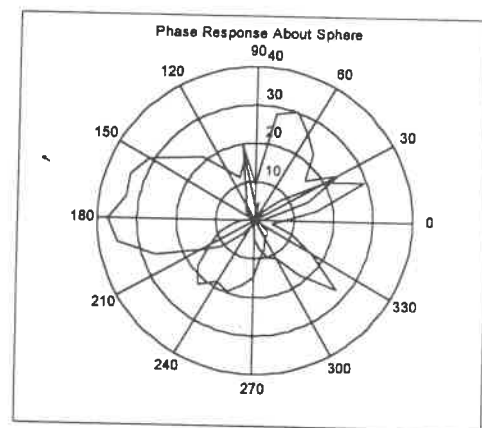


Figure 8: Current Phase Sensitivity

Assuming that the change in sensitivity varies harmonically with the orientation of contact, an ideal response in polar coordinates is shown in figure 7. This shows the expected amplitude from each sensor as a function of the angle of contact between probe and specimen. Figure 8 shows the output from both sensors using the first generation probe at a 200 mN, applied load. The experimental data, though noisy, is encouraging. It clearly shows that each axis has a change in sensitivity as the direction of constant applied force load is changed about the sphere. Based on our observations, it is suspected that inconsistencies in the data are due primarily to the mounting method used for this first prototype. It is hoped that the second generation probes with the nodal mounting should reveal a better two-dimensional characterisation. If successful, a three-dimensional characterisation will be undertaken over the following year.

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Development of A Virtual Gauging System for Cylindrical Coordinate Measuring Machines

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Introduction

Cylindrical coordinate measuring machines are widely used by the world's major reciprocating engine manufacturers and automobile makers. They are mainly used for the roundness measurement of crankshafts of reciprocating engines. Other parameters such as cylindricity and straightness of the journals and parallelism among the journals are also measured during one set-up. Adcole-1200 gauges are a type of specialized precision cylindrical coordinate measuring machines, which adopt the modern laser interferometry technology to automatically monitor the mechanical movement of the carriage and the follower on which three interferometers are mounted. The machine can achieve a resolution of 0.08 micron and measure 1440 data points per revolution within a few seconds.

Crankshafts are key elements of reciprocating engines, and their accuracy greatly affects the engine reliability, balance, fuel economy, and emissions. Now higher quality, greater reliability, and lower noise, vibration and harshness for engines are becoming more and more demanding. These require that the crankshafts be manufactured to greater precision. Adcole-1200 cylindrical coordinate measuring machines play an important role in the manufacturing process of crankshafts. The measuring machines detect the deviation of manufacturing process and ensure high quality of the manufactured crankshafts. The gauge maker's rule "Ten to One" states that the tolerance of the measuring machines should be ten times better than the tolerance of the parts to be inspected. In order to meet the ever-growing demand of high accuracy for the measuring machines, it is necessary to identify the underlying error sources involved in the measurement processes. Kunzmann *et al* have proposed the concept of virtual CMM to evaluate the CMM performance[1], and Frey *et al* have used the virtual machining for machining variation analysis[2][3]. Researchers at the National Institute of Standards and Technology (NIST) are also working on the estimation of CMM measurement uncertainty using Virtual CMM approach[4]. At Adcole Corporation, a prototype of virtual gauging system (Adcole-1200VGS) is developed mainly for the purpose of error identification. In the virtual gauging system, the geometrical relationship are mathematically modeled among the mechanical and optical components of the Adcole-1200 cylindrical coordinate measuring machines.

Virtual Gauging System

Figure 1 is a sketch showing the main structure of Adcole-1200 cylindrical coordinate measuring machines. It mainly consists of carriage, follower, mirror, three interferometers and three laser beams, angle encoder, and head-stock and tail-stock. The crankshafts are installed between the head-stock and the tail-stock of the measuring machines. During the measurement process, the crankshafts are rotated by the head-stock which is driven by the precision spindle. The rotation of the crankshafts causes the follower movement perpendicular to the axis of rotation. The position of the follower when the follower is in contact with the journal surfaces of the crankshafts is precisely monitored by three laser beams and angle encoder. For each revolution, 1440 data points are taken for the roundness evaluation of the crankshaft journals. The vertical movement of the carriage is monitored by another laser beam (not shown in Figure 1).

The principle of the virtual gauging system is to mathematically model the follower geometry and motion, the part geometry and motion, and the relationship among them (Figure 2). In the case of crankshaft measurement, both the part geometry and follower geometry are cylindrical surfaces. Form errors can be added to the ideal cylindrical geometry to obtain the real part geometry and follower geometry. Once the part and follower geometric surfaces are created, the part is rotated and the corresponding follower position is determined. Based on these information, the laser readings are mathematically obtained and then converted to the coordinate data of the sample points on the part surface. Finally, the data points are processed and the roundness value is obtained.

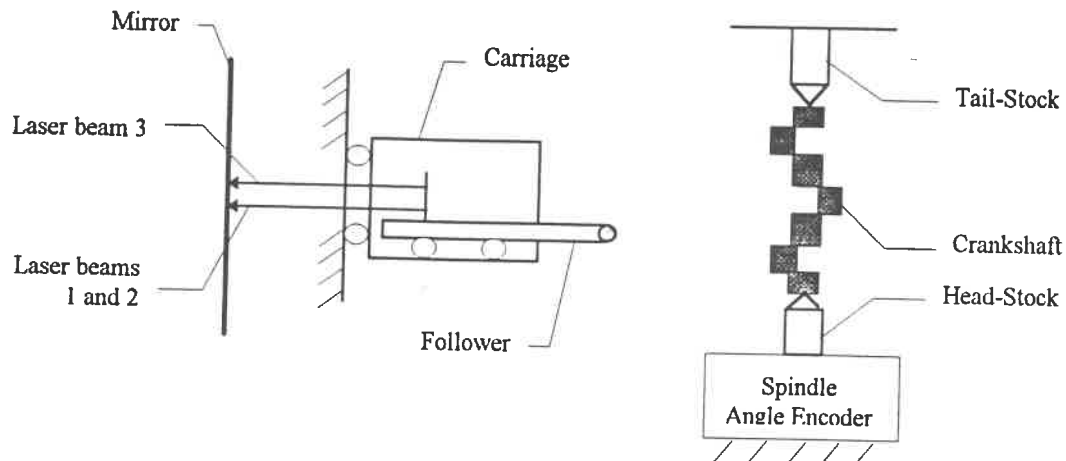


Figure 1 Sketch of Adcole 1200 Cylindrical Coordinate Measuring Machine

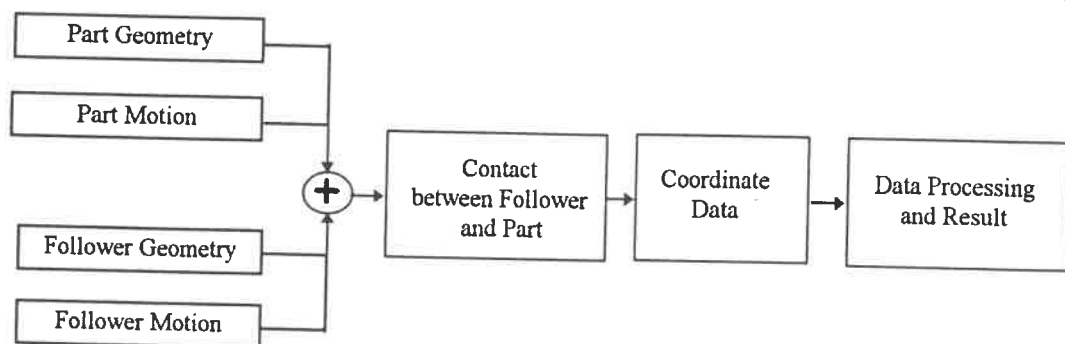


Figure 2 Principle of Virtual Gauging System

Examples

We present three examples of error sources in the crankshaft roundness measurement using cylindrical coordinate measuring machines. Each example corresponds to one type of error source. The error sources presented here are the errors due to part axis tilt, follower straightness, and the spindle error motion. These error sources are created in the virtual gauging system and the roundness error caused by these error sources are obtained by the simulation using the virtual gauging system. The roundness error obtained by the virtual gauging system is also compared with that of the theoretical analysis. The comparison shows that the result from the virtual gauging system is consistent with that of the theoretical analysis.

Error due to Part Axis Tilt

We know that when the axis of a cylindrical part is tilted, the cross-section of the part perpendicular to the axis of rotation is an ellipse instead of a circle. The peak-to-valley value of the roundness error caused by the axis tilt is

$$E_{1,pv} = R(\sec \alpha - 1) \approx \frac{R\alpha^2}{2}, \quad (1)$$

where, the R is the sum of the follower and part radii, and α is the tilt angle of the part axis. Figure 3 gives the simulation results using the virtual gauging system, and the comparison between the simulation and the theoretical analysis of Equation (1). The horizontal axis is the part axis tilt angle α , and the vertical axis is the roundness

error. It is seen that the results agree very well. The largest difference between the two methods is at tilt angle α of 1 degree, which is $0.19\mu\text{m}$ and accounts for only 4% of the theoretical value.

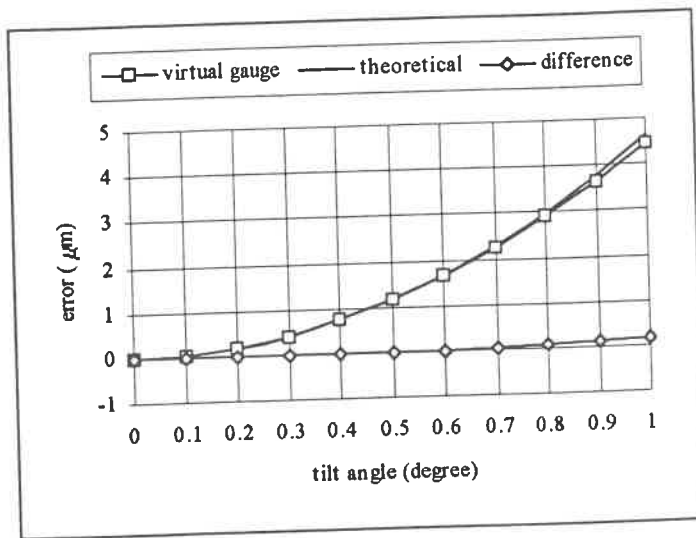


Figure 3. Results due to the part axis tilt ($R=30.48\text{ mm}$)

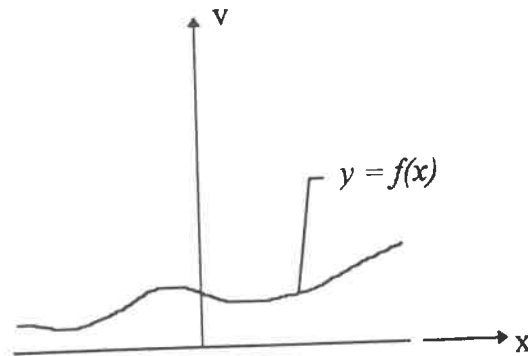


Figure 4. Follower Straightness

Error due to Follower Straightness

In general, the follower straightness can be mathematically expressed as $y = f(x)$, as is shown in Figure 4. For a perfectly straight follower horizontally oriented, the mathematical expression is $y = C$ (C is a constant). It is seen that if the journal to be measured is concentric to the axis of rotation (the case of main journals), the effect of follower straightness error on the roundness measurement is negligible. However, when the journals to be measured are pin journals, the follower straightness becomes very important in the roundness measurement. Assuming that the follower straightness can be mathematically expressed as

$$y = ax^2, \quad (2)$$

where a is a constant, it is easy to show that the peak-to-valley value of the roundness error caused by the follower straightness is as follows:

$$E_{2,pv} = ae^2, \quad (3)$$

where e is the eccentricity of the journal to be measured. When $a=7.75e-7\text{ mm}^{-1}$, the follower straightness would cause a roundness error of $0.5\mu\text{m}$ for $e=25.4\text{ mm}$. Figure 5 shows the simulation and the comparison between the simulation and the theoretical analysis of Equation (3). The results agree so well that the difference is virtually zero (The largest difference is only $0.001\mu\text{m}$).

Error due to spindle error motion

In the roundness measurement using Adcole-1200 cylindrical coordinate measuring machines, the sensitive direction of spindle radial error motion is in the direction of the follower motion. The spindle errors in the sensitive direction will be mapped to the part surface as if it were roundness errors, while the spindle errors in the non-sensitive direction (perpendicular to the follower motion) have little effects on the roundness measurement. In the simulation, uniformly distributed spindle errors of $[-\varepsilon, \varepsilon]$ are added to the sensitive direction (direction of follower motion), and the uniformly distributed spindle errors of $[-2\varepsilon, 2\varepsilon]$ are added to the non-sensitive direction. It is obvious that the roundness error will be dominated by the spindle error motion in the sensitive direction, and the peak-to-valley value of the roundness error will be

$$E_{3,pv} = 2\varepsilon. \quad (4)$$

Table 1 shows the results from the simulation. The first column is the peak-to-valley value of the random error in the sensitive direction, the second column is the roundness error obtained by the simulation due to the spindle error motion, and the third column gives the relative difference between the simulation results (the second column) and the true spindle error motion in the sensitive direction (the first column). It is observed that the difference is very small (less than 3 percent of the true spindle error in the sensitive direction). The simulation also verifies that the spindle error motion in the non-sensitive direction has little effect on the roundness error.

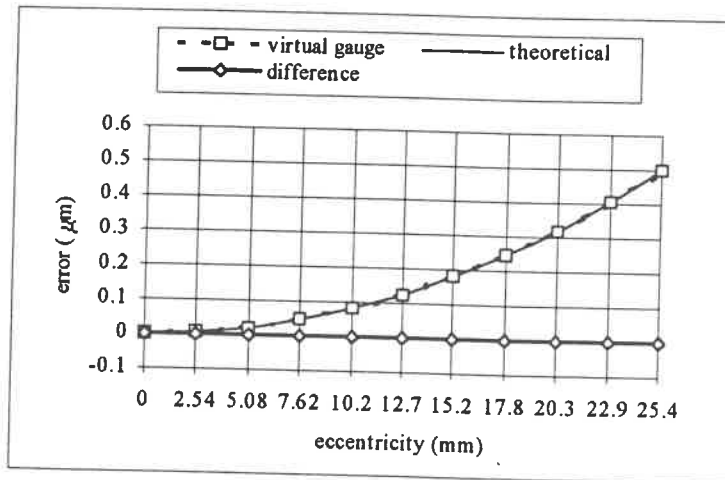


Figure 5. Results due to the follower straightness

Table 1. Results due to the spindle error motion

spindle error 2ε (μm)	roundness error (μm)	relative difference (%)
0.127	0.1305	2.74
0.254	0.2609	2.70
0.508	0.5220	2.75
1.016	1.0437	2.73

Closing Remarks

Due to the complexity of the modern measuring machines, the virtual gauging system is a very useful tool for error identification. By using the virtual gauging system, it is more convenient to manipulate the different error sources, and thus helpful for us to better understand the working mechanism of the sophisticated measurement processes. Finally the underlying error sources in the measurement process can be identified. The identified error sources can then be corrected using the software or hardware methods. In addition to the error identification, the virtual gauging system can also be used to evaluate the overall measurement uncertainty provided that the error sources are properly modeled.

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OPTICAL FIBERS PROFILOMETER FOR DIAMOND MACHINED SURFACES CHARACTERIZATION

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ABSTRACT

The aim of this paper is to present the experimental setup, test procedure and data-processing of a new profilometer, applied to characterize the morphology 2D of a diamond-turned surface. The profilometer is based on an optic-fibers proximity sensor as an optic probe, mounted on a high precision linear stage and having an accuracy less than $0,5 \mu\text{m}$ in z-axis. In order to validate this instrument, a step of $17,80 \mu\text{m}$ was machined with a "square-form" diamond-tool and measured. We found that results obtained with the profilometer were in agreement with measurements carried out on a SIM white-light.

1. INTRODUCTION

Materials surface characterization techniques play an increasing role in a broad range of industrial application, such as surfaces of optical or mechanical components, produced with ultraprecision diamond machining. Among techniques of investigation, optical instruments permit high-accuracy measurements without any surface contact. Laser interferometry¹ is one of the best established methods because of its high resolution and stability of measurements. However, its precision and stability depend on the wavelength of monochromatic light and cannot be obtained for measurements less than $\lambda/4$ due to the fluctuation of the refractive index of the atmosphere. The focus detection technique² involves a complicated combination of optic components and a special design technique. The fiber optic lever sensor is the simplest method to obtain high resolution in z-axis. This paper describes, in section 2, a profilometer based on optical fiber displacement sensor as an optic probe, which is mounted on a high precision linear stage. Optic probe performances consists of a 2 nm resolution in rms on z-axis and linear stage has a precision on its vertical straightness less than $0,5 \mu\text{m}$.³ In section 3, technique of producing smooth surfaces ($R_a \sim 3 \text{ nm}$) and steps (height $\sim 20 \mu\text{m}$) is described and detailed. Section 4 is reserved to present experimental results of a smooth pure aluminum turned surface-profile and the height of a step achieved with a "square-form" diamond tool. We discuss, after, profile-form obtained by profilometer and then validate the height-step measurement by comparison to a scanning white-light interferometric microscope (SIM white-light).

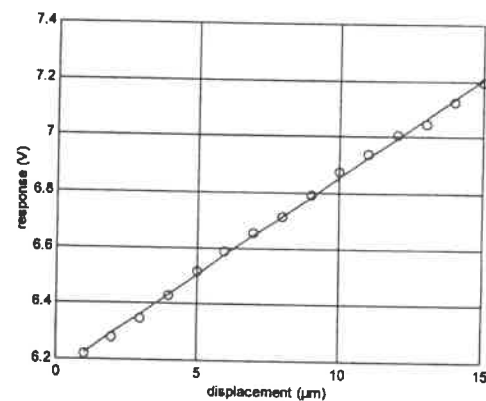
2. OPTIC PROFILOMETER CALIBRATION

Optic probe

The principle of the optical fiber sensor is based on the measurement of the amount of light transmitted from a bundle of fibers and reflected back into another bundle of fibers from a reflective surface, generating a transduction between the topography of surface and the amount of the reflected light power. It is reasonable to expect that the total amount of light reflected depends on type of the material (reflectivity of the surface at the incident beam wavelength and its topography and homogeneity), incident light power, numerical apertures, fiber diameters, and bundle geometry of illuminating-fiber and receiving-fiber. A high radiance AlGaAs, 850 nm LED (HFE 4050) is used as a light source. The LED has a stable forward current to obtain a better optical power output. Light which enters the fiber is transmitted to the other end of the fiber by total internal reflection, with

negligible losses. A PIN photodiode (HFD 3022) is used to transform the light reflected by the surface into an electric signal. The photodiode is connected to a low-input noise operational amplifiers in a current to voltage configuration to achieve the desired resolution. For the calibration, the fiber optic probe was mounted in front of a target mirror moved by a linear low voltage piezoelectric translator (LVPZT) of stroke 15 μm range, to obtain the variation of light intensity. The calibration of displacement is always made by a Michelson laser interferometer (Zygo, Axiom 2/20 : resolution 1,24 nm in position, 0,06 arc sec in angle and 40 nm in straightness). In *Figure 1*, the response of the sensor in function of the displacement of LVPZT is presented for a displacement range of 15 μm . The calibration with the heterodyne Michelson laser interferometer shows that the optical fiber sensor has a stable quasi-linear measuring range from 50 μm up to 200 μm , the sensitivity is 80 mV/ μm and the random noise level has been reduced to less than 10 nm and the noise level indicated by rms is less than 2 nm. Full line represents the least squares fitting and the standard deviation gives an error of 1,5 nm in the determination of the position true value.

Figure 1 Response of the optical fibers sensor versus the displacement of LVPZT. Full line represents the least square fitting. The standard deviation gives an error of 1,5 nm in the determination of the true position value



Precision linear stage

The linear displacement system is mounted on hydrostatic bearings, the guide-ways are made from zerodur with a flatness better than 0,25 μm . The stage is moved by friction drive over a travel of 200 mm and controlled by an optical encoder (*Heidenhain*, LIP 101) with a resolution of 4 nm. A finite element model of the vibration modes of the different elements permitted the design and construction of a stable mechanical system. Indeed, the preliminary results for the characteristics of the stage movement obtained by several measurement series using a Michelson interferometer showed a standard deviation of 0,2 arc sec in pitch, 0,1 arc sec in yaw and 250 nm in straightness. For example, *Figure 2* shows experimental setup and values of the standard deviation of vertical straightness versus the displacement of the high precision linear stage over a travel of 200 mm, it is around 250 nm with a maximum deviation from a straight line trajectory of 0,8 μm .

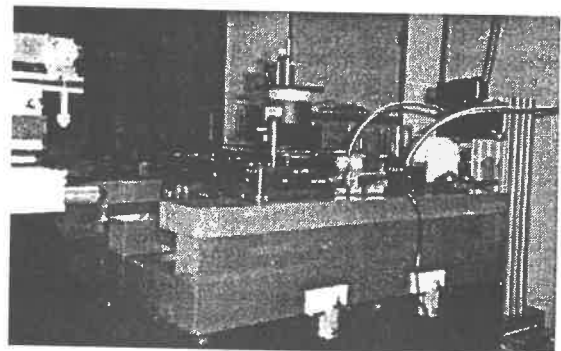
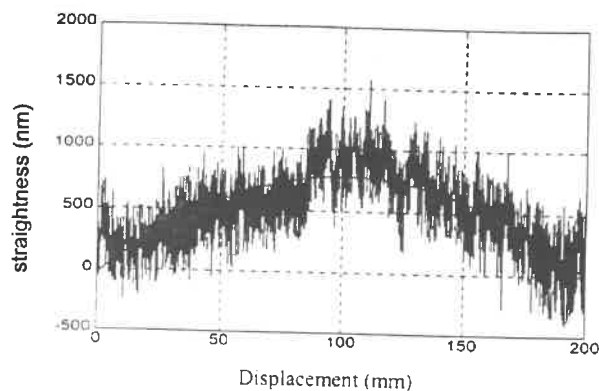


Figure 2 Straightness (in nm) versus the displacement of the stage (in mm)

3. ULTRAPRECISION MACHINING

Machine Description

A precision machine consists of a very stiff and stable (thermally and mechanically) machine base, a 1,5 ton-natural granite base, that operate under closely controlled environmental conditions. To complete the machine, components added to this base include hydrostatic-bearing precision slide-ways (straightness $< 0,3 \mu\text{m}$), magnetic-bearing spindle (with active control), accurate-positioning tool-holder and control processors to guide the machine motions. Displacements are controlled by two optical encoders (*Heidenhain*, LIP 101) with 4 nm resolution and require support computer numerically controlled (CNC) facility. This has normally the capability to produce the smooth components to shape accuracy of approximately 1 to 3 μm .⁴ External vibrations are reduced by supporting the machine on aerostatics-damping. The machine is housed in laboratory equipped with controlled-system maintaining constant temperature, with low humidity and high cleanliness.

Diamond Surfacing

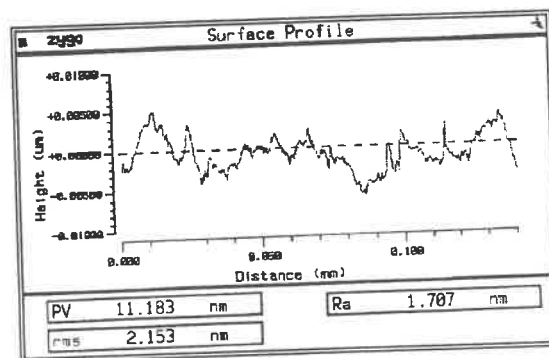
Unlike any other category, single crystal diamond, by its chemical and mechanical properties, permit to obtain not only very sharp edges but very regular edge-shape with, relatively reduced rate of wear. Since ultra-precision machining tenet being that the work-piece shape will be an exact copy of the combination of the tool shape and tool path, then an important attention is accorded to both while machining.

With single-point diamond turning (SPDT), achievable roughness on aluminum or its alloys reaches 2 to 4 nm, with optimal cutting conditions. Even under normal conditions, an appropriate diamond tool offers not less than 10 nm in R_a and less than 1 μm in *peak to valley*.⁵ This technique allows a large range of attainable surfaces, such as flat or curved surfaces, cylindrical and conical surfaces. *Figure 3* shows a profile of a pure aluminum diamond-machined surface, scanned by a SIM white-light (Zygo, New View 200 : resolution 0,1 nm on z-axis). The typical cutting conditions are shown below :

Typical cutting conditions

material	Pure aluminum
type of diamond	natural single-crystal
spindle speed	1200 rpm
feed rate	5 μm / rev
coolant	HYPERZ fluid type OS

Figure 3 Surface profile expressed with roughness parameters (SIM measurement with 50x lens)



Step Machining

To enable producing parts with accurate steps, specific kind of tools was used. Generally, nose radius varies from a half to ten millimetres,⁶ in order to achieve high degree of roughness and shape accuracy. This minimize the friction zone between the tool edge and the machined surface and ensure an accurate cutting.

Using a "square-form" tool with an infinite edge-radius, it was possible to machine steps at different heights and with regular and edged shape. The cutting depth corresponds to the height of the step, that means 20 μm for the step machined to validate the profilometer measurements. It was observed under interferometer microscope that mirror finish wasn't achievable as with single-point diamond tool ; the edge radius is so important that the tool is assimilated to a straight-edged tool.

4. EXPERIMENT AND RESULTS

Calibration

Figure 4 represents a photograph of our optical fibers profilometer, which permits measurements of a large diameter surface-profile and has a vertical dynamic range of 150 μm . Lateral resolution is poor and depends on the emitting optical fiber diameter (200 μm in our case). To increase this resolution, a standard 9/125 monomode optical fiber will be used.

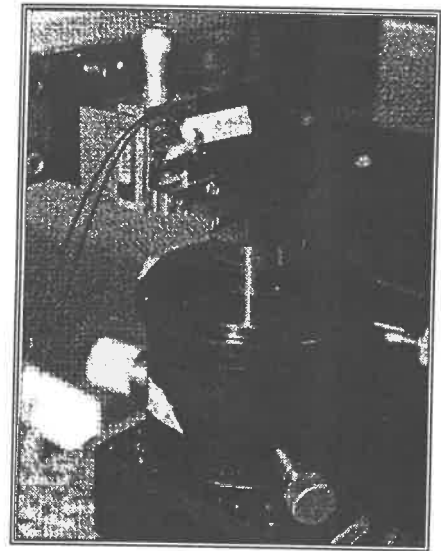


Figure 4 Optical fibers profilometer : analyzed surface is mounted on a natural granite support and tilt-adjusted by means of an adequate plate. Here, the optic probe is in displacement while the part is immobile

Figure 5a shows a Fizeau-interferogram carried out on a reference parallel faces glass-plate of 100 mm in diameter, polished at the *Optic Institute of Orsay University (IOTA)*, results display a flatness of $\lambda/10$. Profile of the same reference-plate was also measured with the profilometer over one radius. Observing Figure 5b, we can distinguish, first, the repeatability of measurement in both ways and second, a flatness less than 0,2 μm .



Figure 5a Fizeau-interferogram of the reference plate

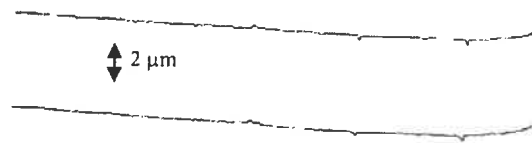


Figure 5b Reference plate profile measured in both directions

Step measurement

We, first, accorded attention to surface smoothness and remarked a high PV ($\sim 1,5 \mu\text{m}$), this was predictable when using a « square-form » tool ; in this case, machining changed from single-point diamond cutting to very classical diamond turning. Our purpose being a regular step machining, the result was evaluated by the SIM white-light and measurement is presented in Figure 6a, the height value was 17,80 μm . The SIM white-light was already calibrated by a NIST step height standard. The certificate of calibration indicates a value of $(1,800 \pm 0.011) \mu\text{m}$.

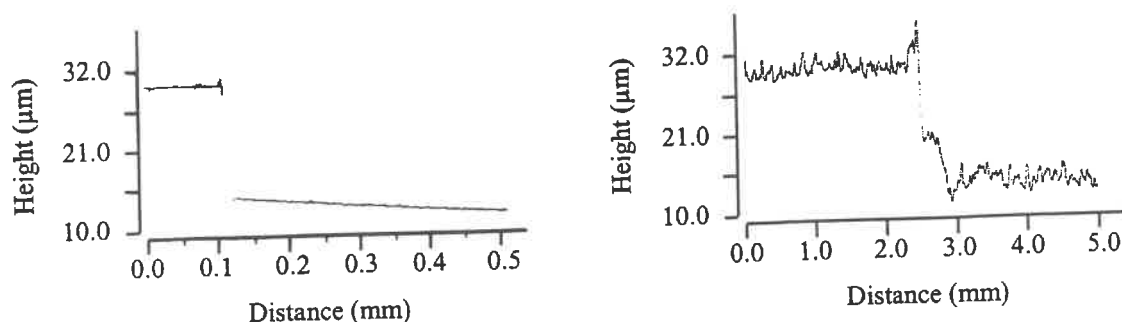


Figure 6a & 6b First curve expresses the SIM white-light measurement of the step (with a 10x lens), which was evaluated to $17,80 \mu\text{m}$; the second represents profile given by the profilometer, the height is $17,75 \mu\text{m}$ (between the two figures, and on horizontal axis, scale was multiplied by 10 for the second)

Carried out with the profilometer, measurement in *Figure 6b* shows humming on its linear parts (i.e. the two levels of the step); these undulations represent low spatial frequencies (flatness) and are about $2,5 \mu\text{m}$ and the step height was $17,75 \mu\text{m}$. At the discontinuity between levels, appears an optic phenomenon associated to profilometer-design. Indeed, the peak observed at the higher level and the valley at the lower one are related to the dimension of the luminous spot scanning the surface texture. This is a mean for the measurement of the profilometer lateral resolution. In our case, it is about $400 \mu\text{m}$.

5. CONCLUSION

The proposed optical fiber profilometer with submicronic vertical accuracy was built to measure morphology of relatively smooth material surfaces (form and flatness). To cover the domain of surface vertical topographic measurements, the profilometer was calibrated by an heterodyne Michelson laser interferometer and has a stable vertical quasi-linear measuring range up to $150 \mu\text{m}$. We have calibrated this profilometer by means of a reference plate and measured a known height of an ultraprecision diamond-machined step. Measurements were in accordance with results given by a SIM white-light. A future study will be concerned with improvement of the lateral resolution using emitting optical fibers with low diameter.

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Solid-avoiding and Richardson Tiling in Surface Metrology

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Introduction

The objective of this paper is to compare two tiling methods of evaluating measured profiles: the traditional Richardson, coastline or compass, method (Brown and Savary 1991, Brown et al., 1996) and a new, solid-avoiding tiling algorithm for evaluating measured surfaces.

Currently the technology for measuring surfaces and for storing and manipulating measured profiles and surfaces exceeds the algorithms available for evaluating and characterizing them. The need for algorithms that do a better job of evaluation and characterization is indicated by the difficulty in establishing functional correlations between surface performance and measurements or between surface measurements and the processes that created them, e.g., manufacturing, wear and fracture.

Most measured profiles and surfaces are characterized by parameters that were developed for use with contact profilers and relatively simple electronics, such as, peak-to-valley roughness, average roughness (Ra) and root mean square roughness (Rq). It has been shown that the information about measured surface contained in the parameter can be calculated (Brown and Hoch 1997), and the parameters can be evaluated on their information content. Furthermore these parameters are calculated after filtering that markedly changes the nature of the profile (Cohen et al. 1997).

Attempts to derive more information from measured surfaces include Fourier analysis, although when compared with length-scale fractal analysis, a kind of 2D-tiling, fractal analysis clearly is a better indicator of the character of the instrument that measured the profile (Malburg 1997).

Length-scale analysis for use on measured profiles (or on measured surfaces considered as a series of profiles) and area-scale analysis have been used in surface characterization and engineering design (Brown et al. 1997, Brown et al. 1996, Brown 1994). The traditional tiling algorithm is based on Richardson's method and fixes the ends of tiling elements, i.e., line segments, to the profile (or points linearly interpolated from measured profile points). This method allows the tiles to cut through the solid, which they will generally do, unless they are tiling a concave region. While the traditional method is convenient computationally, and has clear physical interpretations in many applications, it lacks naturalness in contact applications, and it cannot distinguish a surface from its mirror image. The new method seeks to place tiles on top of the surface so that they do not intersect the surface. The new methods are demonstrated on a profile measured from a turned surface and on a measured honed surface.

Methods

Both tiling algorithms repeatedly measure the length of a measured profile by virtually tiling the profile with line segments. The length of the line segment or tile, is the scale of the measurement or observation, and it is exactly the same within each repetition. With each successive repetition the length, or scale, of the tile is varied so that a significant range of scales is covered. The measured length is compared with the projected length to determine the relative length, as a kind of normalization. The log of the relative length is plotted versus the log of the scale, or tile length and this plot can be used to characterize the texture.

At sufficiently large scales on nominally flat surfaces the relative lengths are approximately equal to one, as the tiles are large enough to be insensitive to the topographic details. At these scales the surface is essentially smooth. At some sufficiently small scale the relative length increases significantly above one, and the surface appears rough. The transition between these smooth and rough regions is the smooth-rough crossover (SRC). The SRC in this study is the scale that corresponds to the relative length that is 10% of the log greatest relative length measured. Below the SRC the surface can often be characterized by the slope of the log-log length-scale plot, which is indicative of the complexity of the profile and can be used to calculate the fractal dimension (Mandelbrot 1984, Brown and Savary 1991). To characterize the complexity we use the length-scale

fractal complexity (L_{sfc}), which is -1000 times the slope of the region of the plot covering an order of magnitude in scale with the steepest slope.

In the classical tiling method, originally developed by Richardson for measuring coastlines and borders (Mandelbrot 1984), the end points of the tiles essentially rest on the profile (Brown and Savary 1991). The tiles can intersect the profile, without regard for whether or not they traverse the solid or void (Fig 1a).

Fig. 1a. A schematic of Richardson tiling.



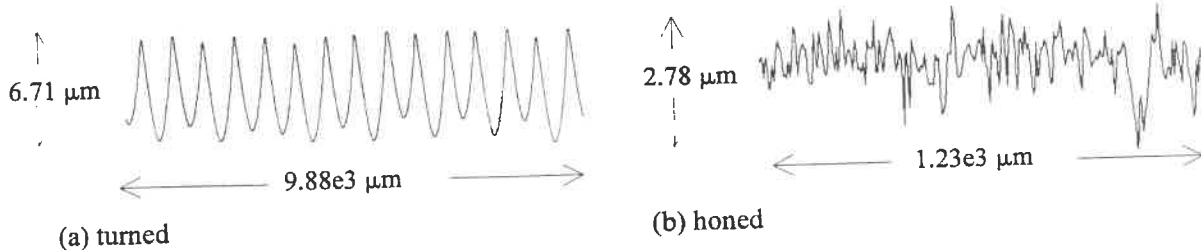
Fig. 1b. A schematic of solid-avoiding tiling.



In the solid-avoiding method, which has recently been developed, the tiles must not traverse the solid side of the profile. The endpoints of the tiles are not constrained to lie on the profile. The tiles essentially lie on top of the profile, supported by two points (Fig 1b).

Two surfaces are examined: a turned surface, and a honed surface. The turned surface is measured with a Federal Products Surfanalyzer 5000 contact profiling device (Providence, RI). The measured profiles consisted of approximately 7600 elevations at a sampling interval of about $1.3\mu\text{m}$. A section of a profile is shown in Fig 2. A section of a honed surface, $915 \times 1230\mu\text{m}$, was measured with an interferometric microscope. The measurement consists of an array of 236×368 elevations. The measured surface for this analysis was considered as 236 separate, measured profiles, $1230\mu\text{m}$ long with sampling intervals of about $3.37\mu\text{m}$. Both surfaces were inverted using software and are analyzed in the as-measured and inverted condition.

Fig. 2. Portions of the profiles from the surfaces.

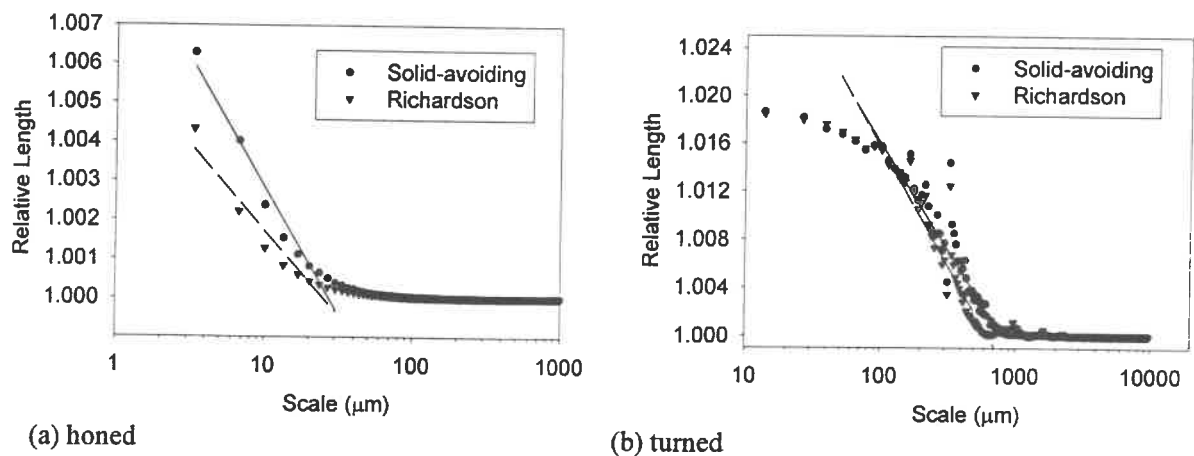


Results and Discussion

The length-scale plots (Fig. 3) and the characterization parameters (Table 1) are used to compare the Richardson and solid-avoiding tiling methods and the extent of the effectiveness of the solid-avoiding tiling for distinguishing a profile and its inverse (Fig. 4).

Fig. 3 shows that there is a clear difference between the two tiling algorithms and that this difference depends on the kind of surface examined. The Richardson plot of the turned surface (Fig. 3a) had relative minimums at even multiples of the feed, which can be used to determine dominant wavelengths (Brown et al. 1996). There is also a spike at half the feed, which is typical of length-scale plots of profiles that have a strong signal at a particular wavelength (Brown and Meacham 1994). At the finer scales the negative slope of the plot decreases, as an indication of the limit of the resolution of the profiling device. Fig. 3b does not show the same decrease in negative slope at fine scales, which may be attributed to not having approached the resolution of the interferometric microscope at these scales. There are two reasons why Fig. 3b is smoother than 3a. The relative lengths on 3a are the means taken over the 236 profiles in the measured surface. Secondly, the honed

Fig. 3. Length-scale plots showing solid-avoiding vs Richardson tiling. The regression lines used for determining the Lsfc are shown.

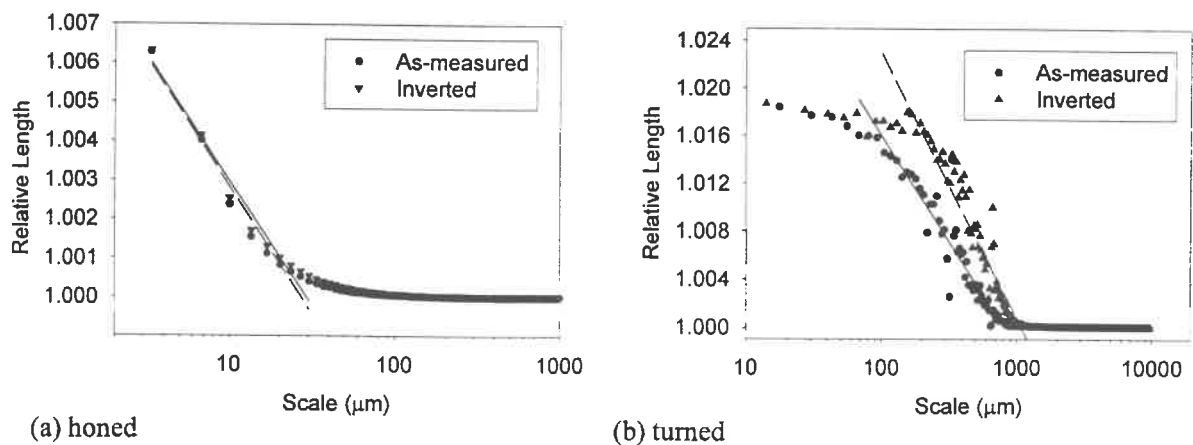


surface is more purely chaotic than the turned surface as it lacks the strong wavelength that is inherent to turning.

On both plots it is evident that the SRC occurs at coarser scales for the solid-avoiding tilings and that in general the solid-avoiding tilings have a larger relative area than do the Richardson tiling. The magnitude of the difference is dependent on the profile, as can be seen by comparing Figures 3a and 3b and by examining Fig. 4.

Fig. 4 shows that the difference between a surface as-measured and its inverse can be detected by the solid-avoiding tiling. Similar plots for Richardson tilings are not shown, because the length-scale plots for as-measured and inverted surfaces are identical. The magnitude of the difference that is

Fig. 4. Length-scale plots calculated using solid-avoiding tiling showing as-measured vs inverted profiles.



evident between the plots in 4a and 4b is indicative of the natures of the surfaces, where the honed surface is relatively similar to its inverse, whereas the turned surface is not.

Table 1 lists the characterization parameters. The conventional parameters, R_a , etc., follow the standard definitions (ASME B46 1995) and are only listed for the as-measured surface. The results for the inverted surfaces are identical for average, rms and peak-to-valley roughness. The peak height for the inverted case is equal to the peak-to-valley minus the peak height for the as-measured surface, as expected. The differences in the tiling-based fractal parameters between the two surfaces and between the as-measured conditions and their inverses quantify the differences noted in the plots.

Table 1. Comparisons of characterization parameters

	as-measured					inverted		Tiling Method
	Ra (μm)	Rq (μm)	Rt (μm)	Rp (μm)	Lsfc	SRC (μm)	Lsfc	SRC (μm)
turned	15.9	18.3	66.8	30.0	7.6	547.4	9.51	788.7
					8.4	445.7	8.4	445.7
	ARa (μm)	ARq (μm)	ARt (μm)	ARp (μm)				
honed	0.339	0.461	6.30	3.09	2.83	23.49	2.77	26.86
					1.89	20.14	1.89	20.14

Conclusions

1. Solid-avoiding tiling produces a length-scale plot that shows that it is clearly different than traditional, Richardson tiling.
2. The nature of the difference between solid-avoiding tiling and Richardson tiling depends on the surface.
3. Solid-avoiding tiling can distinguish between a profile as-measured and its inverse, whereas Richardson tiling cannot.

Acknowledgements:

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Master Axis of Rotation for Spindle Measurement

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BACKGROUND

Characterization of spindle performance is an essential step in the proper evaluation of precision machines. The knowledge gained from these evaluations is used to realize the highest precision possible required in today's manufacturing environment. As Professor Tlustý states, this requires information above and beyond the manufacturer's standard information and needs to be found through proper testing.¹

Higher precision requirements demand higher accuracy from the entire machine. J. Bryan et al. states that this includes the straightness and squareness of the machine, the machine's vibration characteristics, and the quality of both the tool and spindle.² The spindle is typically the most difficult of these parameters to correctly and accurately measure for error. Proper testing procedures must be followed in order to identify the specific sources of error within a machine.^{3,4} When done correctly, structural problems with a machine can be correctly separated from spindle problems.

Traditionally, spindle measurements have been performed using an artifact with an appropriate gage arrangement.¹⁻⁴ The classical artifact is a lapped sphere mounted to a spindle and measured with non-contact sensors.⁵ Spheres, however, lack the ability to allow measurement of face error motion. Also, the compliance of the spindle during operation (further referred to as 'compliance at speed') cannot be measured with a standard sphere. In addition, small diameter spheres are sensitive to variations in the location of the gage target on the diameter.⁶ Hence, an alternative to master spheres is discussed in this paper as a tool for spindle measurement.

INTRODUCTION OF TEST SET-UP

This paper investigates the use of a master axis air-bearing spindle as a suitable technique for spindle measurement. The master axis is used as an alternative to a master test ball to measure the error motion of a spindle. The master axis is mounted co-axially with the spindle being tested as shown below in Figure 1.

The master axis allows for a range of spindle tests to be performed at any operating speed up to 15,000 RPM. Compliance at speed and face error motion can be measured in addition to radial, axial, and tilt error motions. Capacitance probes sense the air gap between the gage and bearing's non-rotating stator. The subject spindle is operated at the desired speed, rotating the master axis rotor. The master axis stator is essentially non-rotating but is not constrained in any other direction. Therefore, the error motion of the subject spindle drives the motion of the master axis.

The master axis is a Professional Instruments Company 3R air-bearing spindle equipped with a 1024-count encoder. This bearing exhibits nanometer (sub-microinch) average and asynchronous error. The validity of the master axis is tested on a Moore No. 3 Jig Borer spindle. This spindle is driven with a belt drive and one horsepower motor. The data collection and analysis is performed with Lion Precision capacitance gages and Lion Precision's Spindle Error Analysis Software.

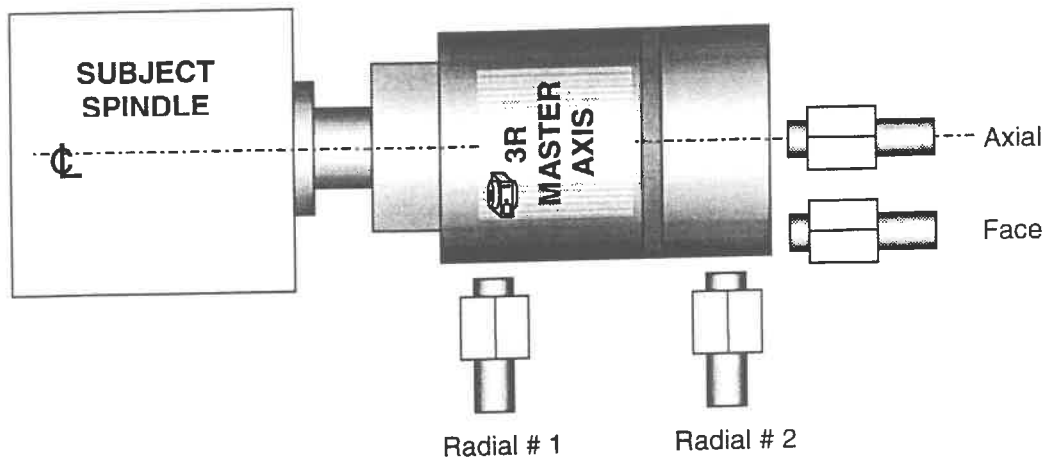
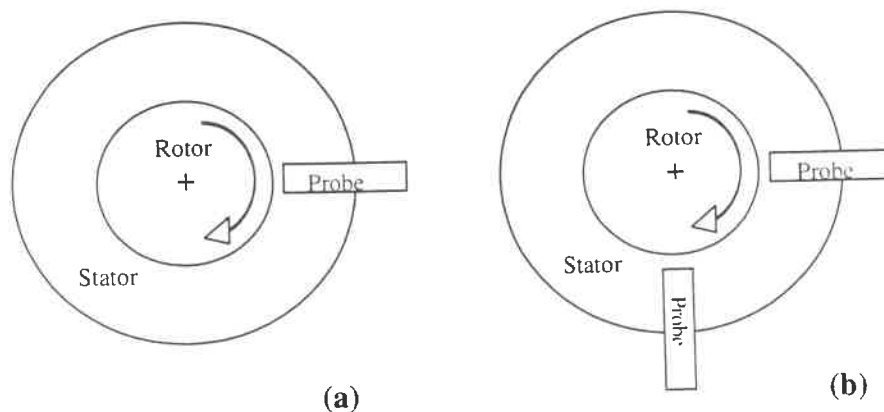


Figure 1: Schematic of Master Axis ⁷

To properly characterize the spindle error motions of the jig borer, a battery of tests are performed using the master axis. First, structural error motion of the machine must be checked in order to avoid misinterpretation of this motion as spindle error. Once, the structural error motion is quantified, the compliance of the spindle is examined to investigate the stiffness of the spindle and its structure. Checking the compliance eliminates the misinterpretation of error motion from a loose joint in the spindle loop (drive, bearings, shaft, master axis attachment, etc.) as error motion.

Finally, the spindle is tested using the proper test for the machine, fixed sensitive direction or rotating sensitive direction. It is important to note that the rotating sensitive error test is the proper spindle test for a jig borer.⁴ The tool is rotated by the borer's spindle while the workpiece is held stationary. This requires two probes measuring spindle error 90° apart from each other. Figure 2 illustrates this concept.



**Figure 2: (a) Fixed Sensitive Direction Measurement (eg. turning)
(b) Rotating Sensitive Direction Measurement (eg. jig boring)**

RESULTS

Before attempting to measure spindle error, structural error motion is measured using two probes aligned 90° apart pointed at the quill of the jig borer. The motor is turned on but the drive is disengaged, therefore the spindle is not turning. This shows the effect of the motor vibration on the error motion of the quill. Figure 3 shows the resulting structural error motion for 16 revolutions of the motor.

It should be noted that the structural error motion in the X or transverse direction is almost 3 times more than in the Y or cross-feed direction. This is expected because the quill design of this machine has higher stiffness in the cross feed direction (Y) than in the transverse direction (X). This error motion was deemed unacceptable and the motor pulley was balanced to decrease structural error motion.

With the structural error motion quantified, the compliance at speed is measured using the two-probe rotating sensitive direction gage arrangement. The master axis is installed on the borer in the position of the tool and operated near 525 RPM. A small scale is used to apply a force on the master axis in the transverse direction (X). The force is incremented from 0 to 44 to 0 Newtons in 8-Newton increments (0-10-0 lbs. in 2 lbs. increments) while the spindle is engaged and running. Figure 4 shows the results of this test, showing a displacement of approximately $12.7 \mu\text{m}$ ($500 \mu\text{-in.}$) at a force of 44 N. (10 lbs.) in the X direction. The displacement in the Y direction remains less than $1.1 \mu\text{m}$ ($40 \mu\text{-in.}$).

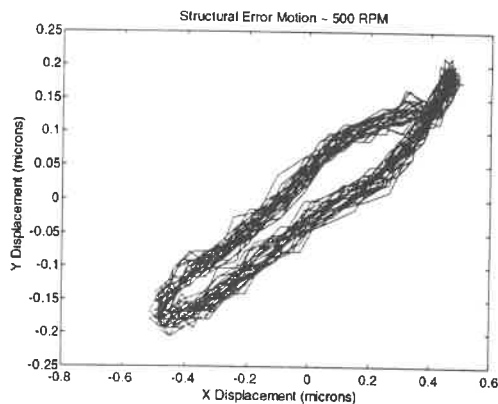


Figure 3: Structural Error Motion

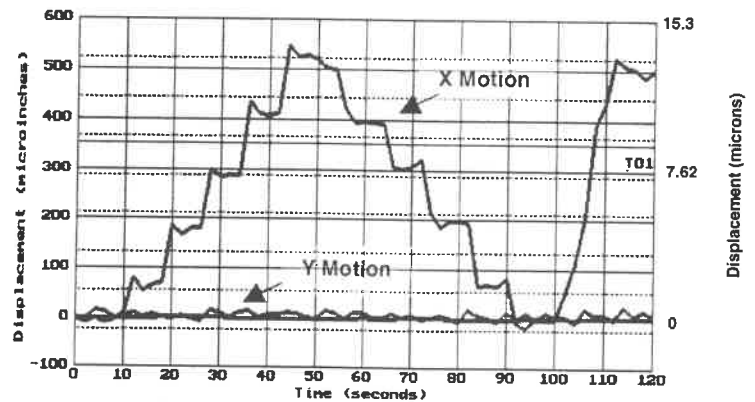


Figure 4: Compliance at Speed

Rotating and fixed sensitive direction measurements are also performed on the spindle. These tests provide the average and asynchronous error of the spindle, which translates into the capability of the spindle to produce accurate parts. The spindle is operated at approximately 530 RPM and data is collected for 32 revolutions with a sampling frequency of 256 samples per revolution. The rotating and fixed sensitive direction results are shown in Figures 5 and 6, respectively.

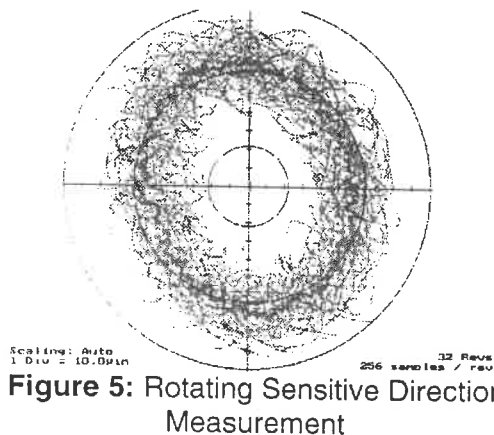


Figure 5: Rotating Sensitive Direction Measurement

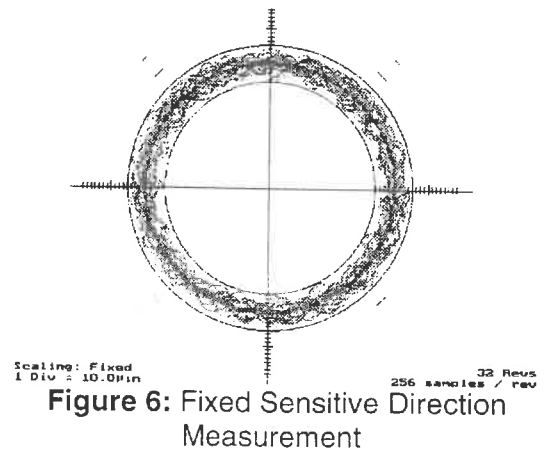


Figure 6: Fixed Sensitive Direction Measurement

The rotating sensitive average error is $0.43 \mu\text{m}$ ($17 \mu\text{-in.}$) with an asynchronous error of $1.7 \mu\text{m}$ ($67 \mu\text{-in.}$). To check the validity of the master axis, a diamond bored hole in an aluminum disc is produced on the same machine and measured using a Federal Formscan. The bored hole measures $1.1 \mu\text{m}$ ($45 \mu\text{-in.}$) out of round. The fixed sensitive average error is $0.18 \mu\text{m}$ ($7 \mu\text{-in.}$) with an asynchronous error of $1.6 \mu\text{m}$ ($65 \mu\text{-in.}$).

This agrees well with the out of roundness measurements of $0.2\text{ }\mu\text{m}$ ($8\text{ }\mu\text{-in.}$) of a diamond turned aluminum cylinder produced on the same machine

Finally, the rotating sensitive direction test is compared to another rotating sensitive test where a constant 22 Newton (5 lbs.) force is applied in the transverse direction with the spindle running at 530 RPM. The resulting polar plot is shown in Figure 8. The polar plot is slightly distorted and shifted in the direction which the force is applied. The center of the circle has shifted approximately $0.30\text{ }\mu\text{m}$ ($12\text{ }\mu\text{-in.}$), which translates into a stiffness of 73.5 MN/m ($420,000\text{ lbs./in.}$) while the spindle is running.

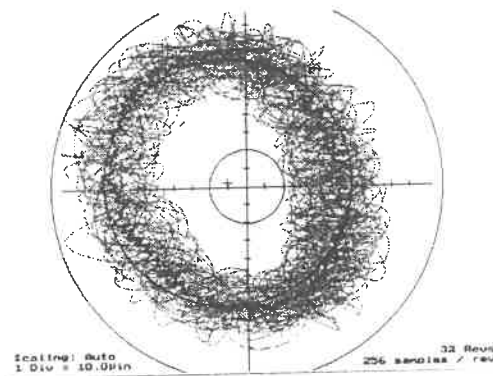


Figure 7: Rotating Sensitive Direction Measurement with 22 N (5-lbs.) Applied Force

CONCLUSIONS

The master axis shows the ability to measure additional spindle characteristics beyond that of a master sphere. Compliance of a spindle at speed can be measured easily by applying a force on the stator of the master axis. Face error and axial error can also be accurately measured against the flat bottom surface of the master axis. Eddy current devices as well as numerous other sensors are compatible with the master axis because the stator is non-rotating. In addition, the encoder also allows for better measurement of rotational position. All of these features make the master axis a feasible alternative to master spheres for spindle measurement.

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3-D Surface Contouring by a Digital Fringe Projection Technique

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1. Introduction

3-D surface contouring techniques have numerous applications in design and manufacturing. For example, they can be used for inspection of industrial parts whose dimensions and geometry need to be checked against their design specifications during or after manufacturing. The same techniques can also be used in reverse engineering where construction of a CAD model from a physical part is required. In recent years, rapid prototyping technology based on layered manufacturing concept has been established which allows for rapid fabrication of physical concept models, functional parts, and tooling directly from CAD models. 3-D surface contouring techniques can help extend the capabilities of current rapid prototyping systems to include building physical parts and tooling from hand-crafted models or parts whose CAD models are nonexistent or missing. They will also help improve the accuracy of the constructed models by introducing in-process or post-process inspection into the rapid prototyping process.

In this paper, we describe a novel 3-D surface contouring technique based on the full-field fringe projection method. The new technique takes full advantage of a novel projection display system to provide the capability of digital fringe projection and phase shifting for significantly improved resolution and accuracy. In the following sections, we describe the principle of this technique and present some experimental results.

2. Digital Fringe Projection

Digital Light Processing (DLP) is a new digital display technology recently developed by Texas Instruments to produce digital video images with high resolution and clarity.¹ Central to DLP is the Digital Micromirror Device (DMD) which is essentially a digital light switch whose surface consists of an array of tiny square aluminum mirrors. Each mirror or pixel of the DMD can be digitally controlled to reflect the incident light into or away from the aperture of the projection lens. Grayscale images are produced by quickly switching the mirrors to digitally pulsewidth modulate the reflected light. Colors are added by the use of a color wheel. Typically, the projection system can be operated in either 8-bit grayscale or 24-bit color mode. Currently available models provide a resolution of either 640 x 480 (VGA) or 800 x 600 (SVGA). The XGA model with 1024 x 768 resolution and models with even higher resolution are being developed.

This new digital projection system can be used to project any grayscale or color images digitally created on a computer screen, including fringe patterns for surface contouring. Because of DLP's high performance, this new fringe projection system can easily produce fringe patterns with high brightness, contrast, and spatial accuracy, which helps in achieving higher resolution and accuracy in 3-D surface contouring. Also, since the fringe patterns are digitally created in a computer, fringe spacing can be easily and quickly changed to adapt to the geometry of the object for optimized contouring.

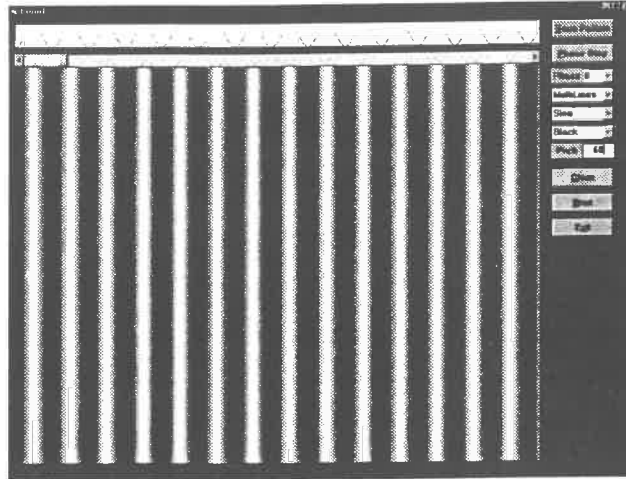


Figure 1 Graphical user interface of the fringe creation software.

To facilitate the creation of fringe patterns, we developed a piece of software in Visual Basic with a graphical user interface as shown in Figure 1. With this software, fringe type, intensity profile, spacing, phase, and color can be easily changed.

3. Digital Phase Shifting

Phase shifting technique has been widely used in optical testing for increased resolution.² Traditionally, phase shifting of fringe patterns has been achieved by using actuators such as PZT to physically move the grating or other optical components in the system. This not only makes the system more complicated, but also introduces new error sources such as actuator nonlinearity into the system. There have been some efforts in using LCD display panels for software-based phase shifting.³ But since LCD display panels provided relatively low image brightness and contrast, only limited results were reported. In this research, we combine the concept of software-based phase shifting with the latest technology in digital projection display to provide a truly digital phase shifting capability. This not only eliminates the need for physically shifting the grating or other optical components, but more importantly, provides digital accuracy of phase shifting which significantly improves contouring accuracy.

Many phase shifting algorithms have been developed each with its own advantages and shortcomings.² In this research, we use the standard three-step algorithm which is the simplest but most prone to phase shifting errors. Following equations represent the intensities of the three phase shifted images:

$$I_1(x, y) = I'(x, y) + I''(x, y)\cos[\phi(x, y) - 2\pi/3], \quad (1)$$

$$I_2(x, y) = I'(x, y) + I''(x, y)\cos[\phi(x, y)], \quad (2)$$

$$I_3(x, y) = I'(x, y) + I''(x, y)\cos[\phi(x, y) + 2\pi/3], \quad (3)$$

where $I'(x, y)$ is the average intensity, $I''(x, y)$ is the intensity modulation, and $\phi(x, y)$ is the phase to be determined. Solving the above three equations simultaneously for $\phi(x, y)$ yields the following solution:

$$\phi(x, y) = \tan^{-1} \left(\sqrt{3} \frac{I_1 - I_3}{2I_2 - I_1 - I_3} \right) \quad (4)$$

With corrections, the above equation provides the so-called modulo 2π phase at each pixel with values between 0 and 2π . Once $\phi(x, y)$ for each pixel is determined, a saw-tooth-like phase-wrapped image can be generated with the highest gray level corresponding to the phase of 2π and the lowest to 0. From this phase-wrapped image, the height information for each pixel of the image can be extracted based on the standard phase unwrapping process.

4. Experimental Results

Figure 2 shows the experimental setup. The fringe patterns, which are sinusoidal with 256 gray levels, are created on a PC screen by using the aforementioned fringe creation software. The fringe patterns are then projected onto the object surface via a DLP projector (LitePro 620 from In Focus, Inc. with 800 x 600 resolution) which is connected to the video port of the PC. A high-resolution black-and-white CCD camera with 2029 x 2048 pixels (4.2i from Kodak) is used for image capture. The camera is interfaced to a Unix workstation via a camera control unit. Once the images are captured and saved, they are sent to the PC for processing. Windows-based software developed in Visual C++ is used for phase-wrapping, phase-unwrapping, and 3-D display. Figure 3 shows the contouring results of a head sculpture by using the proposed technique. Figure 3(a) is a black-and-white photo of the object taken by the same camera at the same position. Figures 3(b)-(d) show the three phase-shifted images of the object. Figure 3(e) is the phase-wrapped image generated by using the three-step algorithm. Figures 3(f) and (g) are the reconstructed 3-D images of the object at different viewing angles. A total of more than 4 million data points were generated but only 1/100 of them was used to create the 3-D images shown in the figure. All data used in creating the 3-D images were raw data without any filtering. Thus the highest spatial resolution was preserved. Judging from the smoothness of the reconstructed 3-D surface contours, we can see that the accuracy of digital phase shifting is indeed very high.

5. Conclusion

We have developed a new 3-D surface contouring system based on a digital fringe projection and phase shifting technique. In this system, fringe patterns are created digitally on a PC and projected to the object surface via a digital projector. Fringe patterns with high brightness, contrast ratio, and spatial accuracy can be easily produced, which enables accurate contouring of large objects as well as objects with low surface reflectivity. For improved contouring resolution, a digital phase shifting technique has been developed in which phase shifting is done by software as compared to actuators in the traditional systems. As a result, phase shifting is more accurate which translates into high accuracy in surface contouring. The new technique should have various applications in the areas of inspection and reverse engineering.

Acknowledgement

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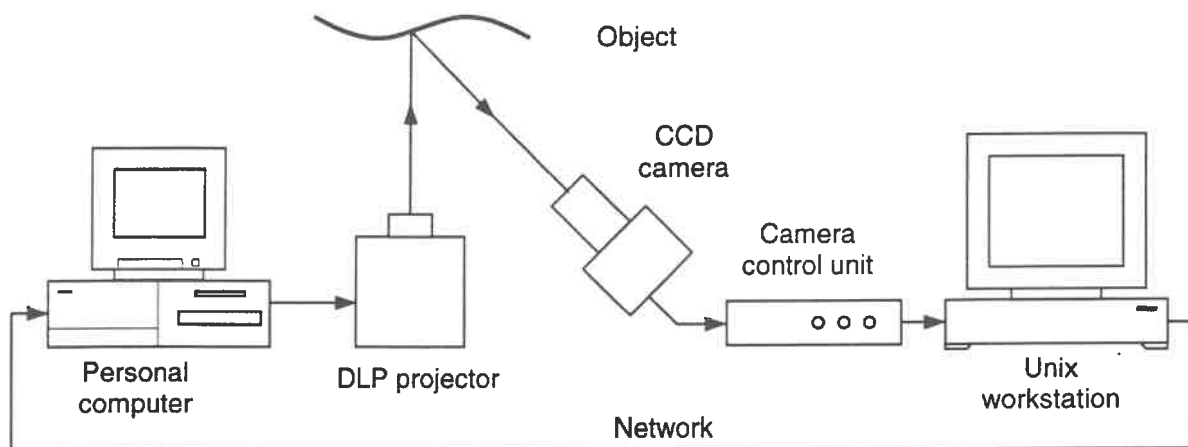


Figure 2 Configuration of the 3-D surface contouring system.

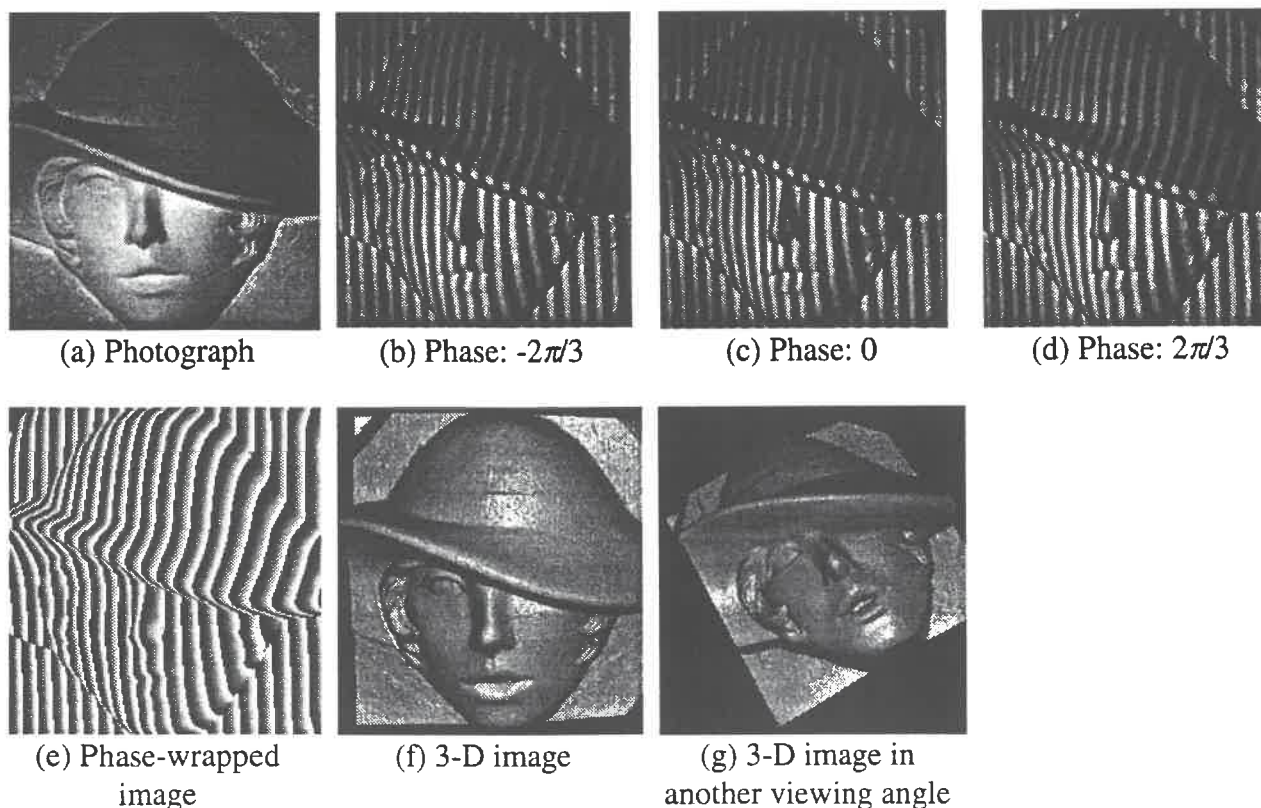


Figure 3 Results of 3-D surface contouring of a head sculpture.

IN-SITU ORTHOGONALITY ALIGNMENT OF PLANE MIRROR INTERFEROMETERS

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Abstract

An *in-situ* calibration method of the orthogonality misalignment of the plane mirror laser interferometers is presented for simultaneous measurement of two orthogonal axis motions of xy-stages. The reversal principle of orthogonality evaluation is realized using a special square artifact that is composed of three optical fiducial marks forming a right angle. The marks are in fact made of concentric-circle gratings, whose two-dimensional positions are precisely located relying on cross-correlation technique. An uncertainty analysis of the proposed method shows that 0.040 arcsec calibration accuracy is achievable when the drift of laser interferometers due to temperature variation is excluded.

Keywords: plane mirror laser interferometers, orthogonality alignment, xy-stages, reversal principle.

INTRODUCTION

Plane mirror laser interferometers use simple straight reflectors, instead of corner cube type retroreflectors. This feature permits concurrent measurements of multiple axis motions with relatively compact optical configurations. In addition, multiple beam pass optics can be readily incorporated to enhance measuring resolution. However, the use of plane mirrors as reflectors requires precise alignment of the mirror normal in reference not only to the interferometer beam but also to the axis of measurement [1,2]. Otherwise, as illustrated in Figure 1, the interferometer beam gets deviated from the axis of measurement causing a typical cosine measurement inaccuracy.

The alignment problem becomes more intricate when xy-motions are to be simultaneously monitored. Two mirror normals of xy-axes should be positioned in line with their corresponding interferometer beams, and they should also be orthogonal to each other. It is consequently a practice to fabricate the xy-mirrors in a single body so that their mutual orthogonality is readily assured on an optical test bench using precision spectrometers and autocollimators [3]. However, the xy-mirrors are often to be made separated for the sake of fabrication especially when large size mirrors are needed. In this case, the orthogonality has to be secured while being installed on the machine, for which a more convenient means than the conventional angle measuring devices becomes necessary.

In this paper, we first present a two-dimensional geometrical alignment error analysis for the xy θ -motions measurement. Then the analysis leads to a quantitative error model predicting that the orthogonality alignment error of the reflectors produces significantly measurement inaccuracy. An *in-situ* assurance method is proposed, in which two-dimensional optical gratings of concentric-circles are used to realize the reversal principle of orthogonality evaluation. Finally, the proposed measurement method is tested to demonstrate that a sub-arcsec measurement uncertainty can be achieved.

STAGE ERRORS

Figure 1 shows the typical setup of plane mirror interferometers for the concurrent measurement of planar xy θ -motions of a stage. Two coordinates systems are defined for analysis

of alignment errors; the mirror coordinates denoted by $\{x_m, y_m, \theta_m\}$ and the measurement coordinates by $\{x, y, \theta\}$. The xy -axes of the measurement coordinates are perfectly orthogonal, along which the $xy\theta$ -motions of the stage are described. On the other hand, the x_m and y_m axes of the mirror coordinates are not orthogonal being set in parallel with the mirror surfaces. For convenience the x -axis is set identical to the x_m -axis. Then the angle φ between the y - and y_m -axes represents the orthogonality error of the mirrors to be concerned with in this investigation.

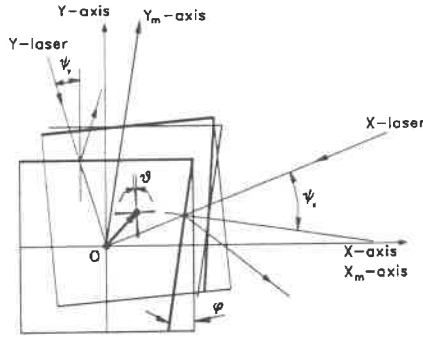


Figure 1. Stage coordinate systems.

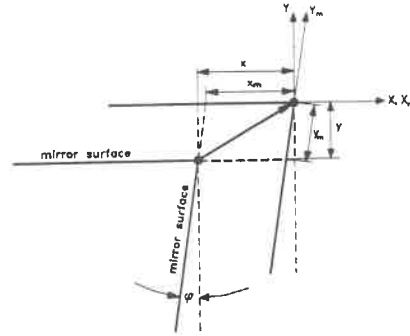


Figure 2. Measurement and mirror coordinates.

Figure 2 illustrates how the measurement coordinates are related to the mirror coordinates. The result is expressed in a matrix form of

$$\begin{bmatrix} x_m \\ y_m \end{bmatrix} = \begin{bmatrix} 1 & -\tan \varphi \\ 0 & 1/\cos \varphi \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix} \quad (1)$$

Now the beam alignment errors are denoted by ψ_x and ψ_y being measured between the interferometer beams and their corresponding mirror normals. These beam misalignments make the axes of measurement be deviated from the mirror normals. Then the measurement output x_i, y_i from the interferometers are related to the x_m, y_m coordinates such as

$$\begin{bmatrix} x_i \\ y_i \end{bmatrix} = \begin{bmatrix} \cos \varphi \cos \psi_x & 0 \\ 0 & \cos \varphi \cos \psi_y \end{bmatrix} \begin{bmatrix} x_m \\ y_m \end{bmatrix}. \quad (2)$$

Combining Equations (1) and (2), and including the θ -motion lead to a generalized error model of

$$\begin{bmatrix} x_i \\ y_i \\ \theta_i \end{bmatrix} = \begin{bmatrix} \cos \varphi \cos \psi_x & -\sin \varphi \cos \psi_x & -y \\ 0 & \cos \psi_y & x \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ \theta \end{bmatrix}. \quad (3)$$

In practice, the amounts of the alignment errors, φ, ψ_x and ψ_y are so small that $\sin \varphi \approx \varphi$ and $\cos \varphi \approx \cos \psi_x \approx \cos \psi_y \approx 1$. Thus the above error model of Equation (3) is approximated as

$$\begin{bmatrix} x_i \\ y_i \\ \theta_i \end{bmatrix} = \begin{bmatrix} 1 & -\varphi & -y \\ 0 & 1 & x \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ \theta \end{bmatrix}. \quad (4)$$

Now it is important to note that the beam alignment errors, ψ_x and ψ_y , induce cosine type

measurement errors which are negligible in most cases. On the other hand, the orthogonality error φ yields a first order measurement inaccuracy that can not be ignored for accurate concurrent measurement of $xy\theta$ -motions. In practice, however, it is not easy to reduce the orthogonality error φ to a negligible order. One practical solution is the real-time correction in that the orthogonality error φ is measured *a priori* and the true xy -motions are compensated for using the relationship of Equation (4). For this, it becomes necessary to have a suitable orthogonality evaluation method that can be implemented *in situ*.

REVERSAL PRINCIPLE

The reversal principle is known as a general means for accurate measurement of part features without reference to externally calibrated artifacts [4,5]. This principle is applicable to the orthogonality evaluation of the xy -stage under consideration. Let us assume there is a square of which out-of-squareness is to be examined. The stage is operated as a xy -coordinate measuring system, on which the square is set with its one side parallel with the x -axis as illustrated in Figure 3(a). Then the stage is moved along the y -axis so that the out-of-squareness γ of the square is traced using an appropriate displacement probe. If the xy -stage has the orthogonality error φ , the out-of-squareness is measured as $\alpha = \gamma - \varphi$. The square is then reversed as in Figure 3(b) and tested again the same way as previous. This time the result is obtained as $\beta = -\gamma - \varphi$. Combining the two results, the orthogonality error of the stage itself is extracted as

$$\varphi = -(\alpha + \beta) / 2 \quad (5)$$

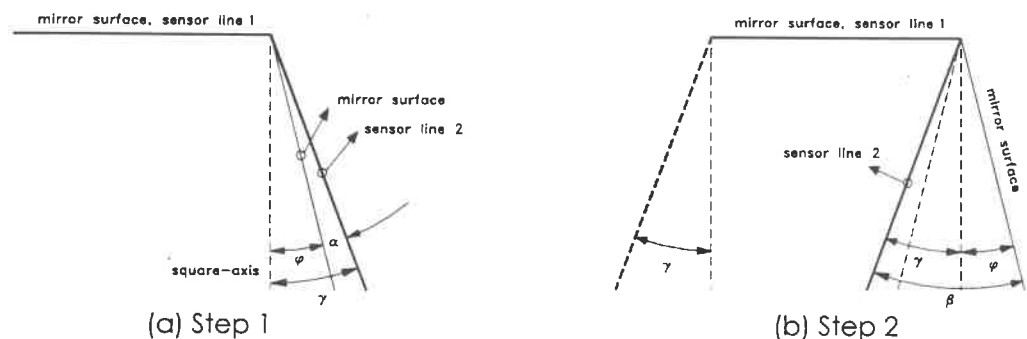
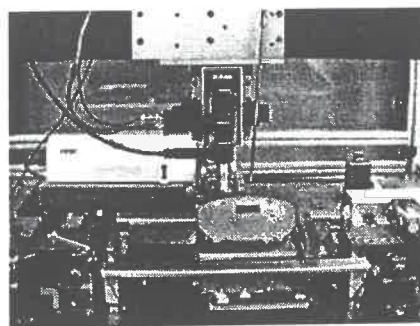
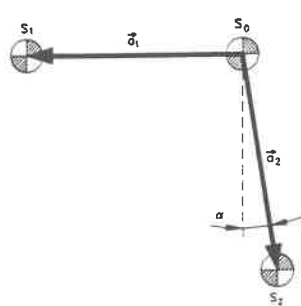
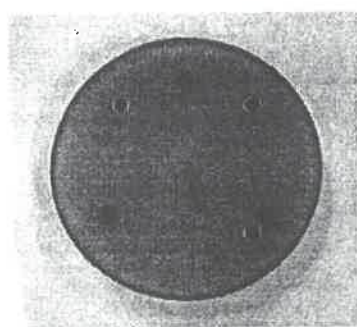


Figure 3. Reversal principle of orthogonality evaluation.

The two-step calibration procedure relying on the reversal principle allows an absolute measurement of the orthogonality of the stage, not being affected by the geometric imperfection of the square. It is however important to note that the overall accuracy of φ depends upon the measurement precision of α and β as conceived from Equation (5). With an aim to achieving a sub-arcsec uncertainty in orthogonality evaluation, a special type square artifact has been prepared in this investigation. As seen in Figure 4, the artifact has three identical fiducial marks, which are positioned to form a nearly right angle that is defined by the two line vectors connecting the marks as depicted in Figure 4(b). The marks are made of optical gratings of concentric circles with 24 μm pitch, whose center is to be precisely monitored by a digital microscope fixed on the z -axis of the stage (Figure 4(c)). The square artifact is made up of the Zerodur material for to minimize thermal deformation due to temperature variation. Further, it is in a cylindrical shape of short height without any holes or apertures in it in order to maintain a complete geometrical symmetry to avoid thermal distortion. A kinematic support of three steels balls holds the artifact on the stage so that no undesirable friction forces due to thermal expansion of the stage is induced.



(a)

(b)

(c)

Figure 4. The square artifact with three optical marks.

CROSS CORRELATION

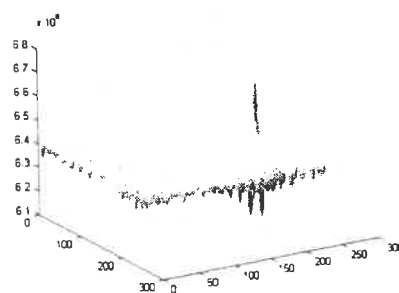
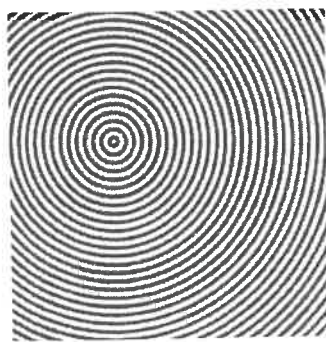
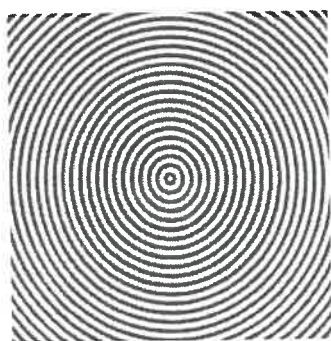
Figure 5(a) shows a typical image of concentric circles of the fiducial marks, which is captured by the digital microscope with a $\times 5$ objective lens. Cross correlation technique is used to precisely detect the center location the concentric circles. For analysis, the reference image intensity $f(x,y)$ is prepared so that the center of the concentric circles coincides with the coordinate center of the CCD image plane of the digital microscope. Let $g(x,y)$ be a target image intensity that is to be located. The first step is to transform $f(x,y)$ and $g(x,y)$ such as

$$F(\xi, \eta) = FFT(f(x, y)) \quad \text{and} \quad G(\xi, \eta) = FFT(g(x, y)) \quad (6)$$

where FFT means the Fourier transform. From its mathematical definition, the cross correlation of $f(x,y)$ and $g(x,y)$ is obtained as

$$r(x, y) = FFT^{-1}(F(\xi, \eta) \times G^*(\xi, \eta)) \quad (7)$$

where $G^*(\xi, \eta)$ denotes the complex conjugate of $G(\xi, \eta)$. Figure 5(c) shows a computed result of $r(x,y)$ as an example in which 256×256 pixel data are processed. The relative position of $g(x,y)$ with respect to $f(x,y)$, i.e., the coordinate center of the CCD image plane, is indicated by the location of the maximum peak of $r(x,y)$. A special peak detection method of Ref. [6] is adopted, in which the peak is found with a sub-pixel resolution using 5 neighboring points about the peak.



(a)

(b)

(c)

Figure 5. Circular grating and cross-correlation result; (a) the reference grating image, (b) a shifted grating image, (c) the cross-correlation result of the images (a) and (b)

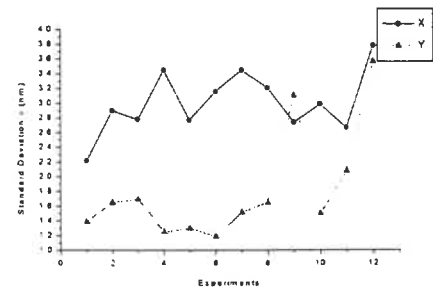
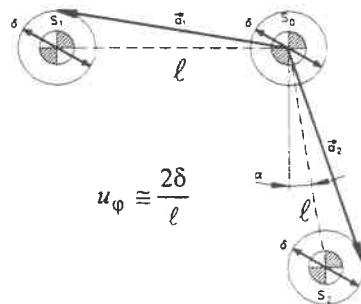
UNCERTAINTY ANALYSIS

To evaluate the performances of the calibration method proposed so far, it is necessary to implement relevant uncertainty analysis. As explained previously, the use of reversal principle provides an advantage that the measurement accuracy of the orthogonality misalignment φ is free from the imperfection of the square artifact. This implies that the square artifact is not required to maintain exact right angle, as long as the angles of α and β of Equation (5) can be accurately determined. Figure 6 illustrates how the uncertainty in the angles α and β can be estimated. Let us assume that the uncertainty in locating the marks is given by δ , then the maximum error in α can be approximated as $u_\alpha \approx 2\delta/\ell$ where ℓ is the nominal distance between the marks. The same estimation can be extended to the angle β as $u_\beta \approx 2\delta/\ell$. Then, from Equation (5), the uncertainty of the orthogonality misalignment φ is finally obtained such as

$$u_\varphi = \frac{1}{2}u_\alpha + \frac{1}{2}u_\beta \quad \text{or} \quad u_\varphi = \frac{2\delta}{\ell} \quad (8)$$

Now the value of δ is to be determined, but its analytical estimation is not an easy task. The reason is that there are so many sources of error to be considered including the optical components constituting the digital microscope, such as the objective lens aberrations, CCD resolution, illumination, and electrical noise in image capturing, as discussed in Ref. [7]. Therefore, δ was determined through experimental approach. Figure 7 shows an experimental result in which the measurement repeatability of the cross-correlation method has been tested. The standard deviation plotted on the ordinate was computed from 100 consecutive measurements performed on an identical concentric-circle grating. This test was repeated 12 times, and the 2σ repeatability was finally obtained as 6.02 nm and 3.66 nm in the x and y directions, respectively. The value of δ is therefore decided as 6.02 nm and this leads to $u_\varphi = 0.04$ arcsec.

Figure 6. Uncertainty of proposed method Figure 7. Repeatability of Cross-correlation



It is important to note that in practice the value of u_φ is affected by the instability of the laser interferometers of the xy-stage since they are involved in measuring the xy-coordinates of the marks along with the local image coordinates of the digital microscope. Hence, if the drift of the laser outputs during calibration is known, it should be added to the uncertainty of δ for accurate prediction of u_φ . In general, the laser drift turns out to be much larger than δ unless environment including temperature is well controlled. The square artifact itself is not deformed by uniform temperature variation, since it has been symmetrically made up in cylindrical shape causing no shear stress. But special care is needed not to induce any temperature gradient within the artifact, especially when reversing it on the stage by hands. Further extensive uncertainty analysis is underway to include all the possible sources of error, so that complete understanding of the

measurement performances of the proposed method will be available.

CONCLUSIONS

We have proposed an *in-situ* calibration method of the orthogonality misalignment of the plane mirror laser interferometers for simultaneous measurement of two orthogonal axis motions of xy-stages. The reversal principle of orthogonality evaluation has been realized using a special square artifact being composed of three optical fiducial marks forming a right angle. An uncertainty analysis of the proposed method shows that 0.040 arcsec calibration accuracy is achievable when environment including temperature variation is well controlled. We now intend to extend this study to a complete calibration of xy-stages including all the geometric errors as attempted in Ref. [11].

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3D volumetric error calibration/correction system for parametric errors in CNC machine tools using a kinematic ball bar

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1. Introduction

Error calibration and correction techniques for CNC machine tools have been classified as very important tasks for users and manufacturers of machine tools in pursuit of quality assurance and maintaining high accuracy during machining operation. There have been several methods for the error calibration/correction system for machine tools, and kinematic ball bar has been focused as an effective tool for multi axis machine tools. Currently, the ball bar system serves as a simple tool for checking contour accuracy, rather than error calibration or error correction. In this paper, we developed the error calibration/correction system using the ball bar measurements, by which the 3D volumetric error components can be efficiently analysed and thus corrected in terms of parameter correction in commercial CNC machine tools. The developed calibration/correction system has been applied to several cases of the CNC machine tools. The error corrected machine tools are retested, and the parametric errors are remarkably reduced, demonstrating the effectiveness of the error correction system.

2. Volumetric Error Calibration

For the volumetric error calibration, a volumetric error model is proposed such that the ball bar measurement data are analysed and thus the parametric error components can be decomposed from the measured data. The proposed volumetric error model can resolve the measured data into about 24 error components such as scale error, squareness error, straightness error, angular error, reversal error, and controller errors.

2.1 Volumetric Error Modeling

Volumetric error equations generally can be derived by the kinematic chain of each machine element when the parametric error components are calibrated, and they are dependent on the kinematic configuration of machine tools.

For a typical column type machining center, the volumetric error equations[1] are,

$$\begin{aligned}\Delta X &= \delta x(X) - \delta x(Y) + Z[-E_y(Y) - \beta_1 - E_y(X)] \\ &\quad + Y[E_z(Y) + E_z(X)] + Y_p[E_z(X) + E_z(Y) - E_z(Z)] + Z_p[-E_y(X) - E_y(Y) + E_y(Z)] \\ \Delta Y &= -\delta y(X) + \delta y(Y) + \delta y(Z) + X[-E_z(X) + \alpha] + Z[Ex(X) + Ex(Y) - \beta_2] \\ &\quad + X_p[-E_z(X) - E_z(Y) + E_z(Z)] + Z_p[Ex(X) + Ex(Y) - Ex(Z)] \\ \Delta Z &= -\delta z(X) - \delta z(Y) + \delta z(Z) + XE_y(X) + Y[-Ex(X) - Ex(Y)] \\ &\quad + X_p[E_y(X) - E_y(Y) - E_y(Z)] + Y_p[-Ex(X) - Ex(Y) + Ex(Z)]\end{aligned}\tag{1}$$

where X, Y, Z are the nominal coordinates from the origin,
 X_p, Y_p, Z_p are the coordinates of the tool offset,
 $dx(X), dy(Y), dz(Z)$ are the Positional error components along X, Y, Z axis,
 $dy(X), dz(X); dx(Y), dz(Y); dx(Z), dy(Z)$ are the straightness error components along X, Y, Z axis,
 $Ex(X), Ey(Y), Ez(Z)$ are the roll error components along the X, Y, Z axis,
and α, β_1, β_2 are the squareness error components in XY, XZ, YZ plane,

respectively.

2.2 Integration of the Volumetric Error Equations and the Ball Bar Measurement

The influence of the volumetric error behaviour of machine tools on the circular error pattern of the ball bar measurement can be considered when equation (1) is applied to the circular error equation[2],

$$\Delta R = (X\Delta X + Y\Delta Y + Z\Delta Z) / R \quad (2)$$

Thus when the volumetric error behaviour is involved for the column type machining center, the circular error of ball bar measurement is,

$$\begin{aligned} \Delta R = & dxx_1(X/R)^2 + dyy_1(Y/R)^2 + dzz_1(Z/R)^2 - 3dxy_1(X/R)(Y/R)^2 + dxz_2(X/R)(Z/R)^2 \\ & - dyx_2(Y/R)(X/R)^2 + dyz_2(Y/R)(Z/R)^2 + dzx_2(Z/R)(X/R)^2 + dzy_2(Z/R)(Y/R)^2 \\ & - Bx / 2(X/R)\text{sign}(dX/dt) - By / 2(Y/R)\text{sign}(dY/dt) - Bz / 2(Z/R)\text{sign}(dZ/dt) \\ & - \beta 1(Z/R)(X/R) + \alpha(X/R)(Y/R) - \beta 2(Y/R)(Z/R) \\ & + Mxy(X/R)(Y/R)\text{Dir} + Myz(Y/R)(Z/R)\text{Dir} + Mzx(Z/R)(X/R)\text{Dir} \end{aligned} \quad (3)$$

where Mxy, Myz, Mzx are the coefficients for the gain mismatch between X-Y axis, Y-Z axis and Z-X axis, respectively

Bx, By, Bz are the backlash error components along X, Y, Z axis,

sign() is the sign function returning the sign of the term inside the bracket, and Dir is a function returning plus(Clockwise) or minus(Counterclockwise) depending on the rotation of kinematic ball bar.

2.3 Least Squares Technique for the Volumetric Error Calibration

When the ball bar measurement data, ΔR , are provided, there are several error components involved as in equation(3). In order to analyse the error components from the ball bar measurement data, the least squares technique has been effectively applied in this paper.

3. Volumetric Error Correction

The decomposed error can now be used for the correction stage, where the correction procedures are performed in order to correct appropriate parameters and variables in the CNC controllers.

The currently available correcting components are: scale errors, reversal errors, straightness errors, squareness errors, controller errors, etc. The correcting procedures are performed via communication networks of the CNC controllers. The corrected machine tools are now tested again for verifying the correction process, and further calibration/correction procedures are followed if necessary.

4. Practical Application

The developed error analysis technique has been practically applied to the column type machining center, which is installed on the shop floor in the Seoul National University.

A kinematic ball bar of 150mm nominal length has been used for the circular error measurement, and fig.1 shows a typical setup for the ball bar measurement. The ball bar has been precalibrated with respect to a calibration fixture made of Zerodour, prior to performing measurement. After the measurement has been performed, the measured data are then

analysed and the error calibration has been performed, and fig.2 shows the developed computer program for the error calibration. Fig.3 shows the ball bar measurement data, and the result of the decomposed error components, desolving 24 parametric error components. After the volumetric error calibration procedures are performed, then the error correction procedures have been performed based on the docomposed parametric error components. The error corrected machine tools are retested, and the parametric errors are remarkably reduced, demonstrating the effectiveness of the error correction system. Fig4. shows the ball bar measurement data after correcting the error components.

5. Conclusion

(1) A computer aided 3D volumetric error calibration/correction system has been developed for the parametric error components, based on the circular error measurement using the kinematic ball bar.

(2) A new approach has been proposed and tested such that the 3D volumetric error model is effectively integrated for the ball bar measurement, thus the 3D volumetric error components have been efficiently calibrated with the ball bar measurement data.

(3) The developed system has been applied to practical cases of machine tools, and demonstrated high efficiency for the calibration/correction of the 3 dimensional error components for machine tools.

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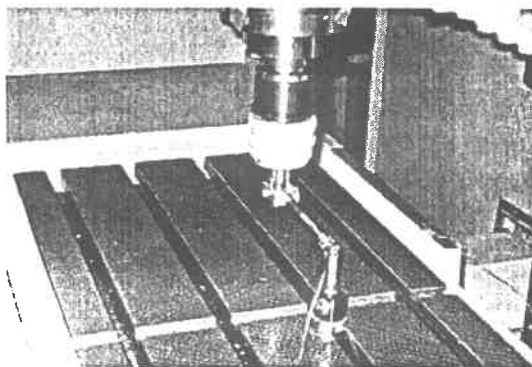


Fig.1 Setup for the ball bar measurement

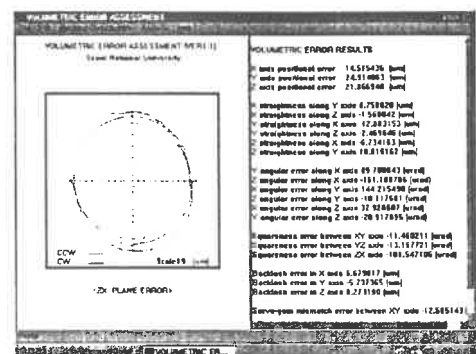


Fig.2 Developed computer program

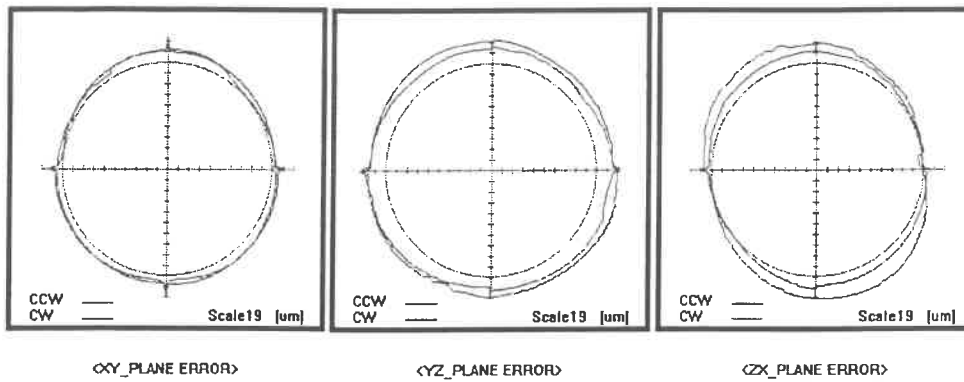


Fig.3 Ball bar measurement plot and the decomposed error components

Error components	Value
X positional error	14.51 [um]
Y positional error	24.91 [um]
Z positional error	21.86 [um]
X straightness along Y axis	0.75 [um]
X straightness along Z axis	-1.56 [um]
Y straightness along X axis	-12.08 [um]
Y straightness along Z axis	-2.46 [um]
Z straightness along X axis	-6.73 [um]
Z straightness along Y axis	10.81 [um]
Y angular error along X axis	89.78 [urad]
Z angular error along X axis	-161.10 [urad]
X angular error along Y axis	144.21 [urad]
Z angular error along Y axis	-10.11 [urad]
X angular error along Z axis	32.92 [urad]
Y angular error along Z axis	-20.91 [urad]
Squareness error between XY axis	-11.46 [urad]
Squareness error between YZ axis	-13.16 [urad]
Squareness error between ZX axis	-101.54 [urad]
Backlash error in X axis	6.67 [um]
Backlash error in Y axis	-5.73 [um]
Backlash error in Z axis	8.27 [um]
Servo-gain mismatch error between XY axis	-12.50 [um]
Servo-gain mismatch error between YZ axis	0.08 [um]
Servo-gain mismatch error between ZX axis	-19.75 [um]

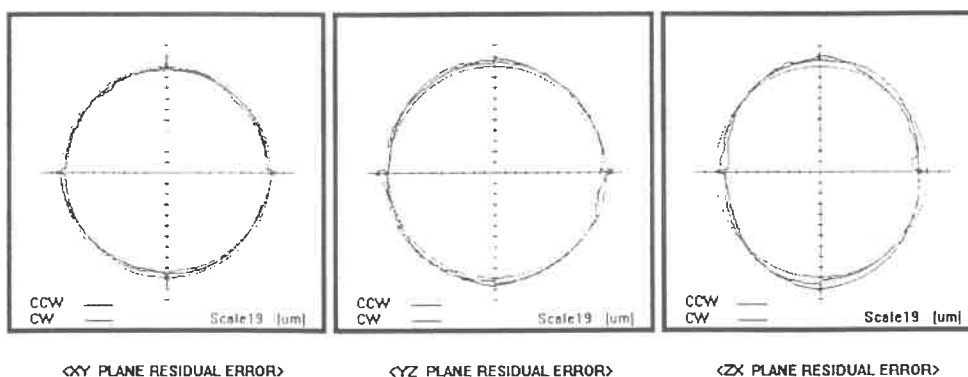


Fig.4 Ball bar measurement plot after correcting the error components

VERTICAL MACHINING CENTER ACCURACY CHARACTERIZATION USING LASER INTERFEROMETER

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ABSTRACT:

This paper presents the results of accuracy characterization of a Vertical Machining Center (VMC) in the form of linear / angular errors and temperature variation using a low powered He-Ne laser (Renishaw®) calibration system along with environmental controller unit. The machine investigated is Cincinnati Milacron Sabre 750 3-axes CNC Vertical Machining Center with Acramatic 2100 CNC open architecture controller. The accuracy of the VMC is characterized in the form of geometric and thermal errors as a function of machine tool nominal axis position, temperature distribution and environmental effect (air temperature, air pressure and relative humidity). The measured temperatures in conjunction with the geometric models are used to predict machine tool geometric and thermal errors. Results show that axis drive motors are the major heat sources. Linear positional accuracy is best when the machine is in cold condition and deteriorates with increasing machine operation time for all 3-axes. X-axis had worst linear displacement accuracy and maximum reversal error among the three axes being tested. X-axis pitch and Y-axis yaw error were highest angular errors of the VMC.

KEY WORDS: CNC machine tool accuracy, laser interferometer, thermal and geometrical errors, error modeling, error prediction.

INTRODUCTION

The accuracy of machined part dimensions depends upon the positional accuracy of the cutting tool relative to the part being machined. Therefore the accuracy of the machine tool used to produce the part is often the limiting factor to obtaining the highest accuracy and part quality. The accuracy of the machine tool is primarily effected by the geometric errors caused by mechanical-geometrical imperfections, misalignments and wear of the elements of the machine structure, by the non-uniform thermal expansion of the machine structure and static/dynamic load induced errors. Tlustý [1] clearly indicated the importance of machine geometry, thermal effects and machine loading. Researchers have extensively performed studies on machine tool accuracy after 70's. Techniques for the automatic compensation of alignment errors and volumetric accuracy modeling of machine tool positioning have been reported [2-4]. Donmez et-al [5] presented a general methodology for predicting and compensating for machine tool errors using homogenous matrix transformations. Kurtoglu [6] published a paper that combined much of the previous work and demonstrated the efficiency of the correction on machined parts. Duffie [7] used kinematics in modeling machine tool errors for compensation. Dorndorf et-al [8] developed a model for quasi-static errors and its use as the basis for an optimal error budgeting of the errors. Shin [9] et-al proposed seven different procedures for the characterization of CNC machine tools. Recently many researchers have started to use Artificial Neural Networks for error prediction and compensation [10]. Most of the error compensation

methods involve the mapping of machine errors and correcting for the effect of the errors. Generally, hardware improvements are necessary to maintain thermal equilibrium throughout the whole machining process [11-12]. Usually this requires environmental control, high performance coolant systems and expensive low friction bearings and drive-lines. However, due to physical limitations, solely production and design techniques can not improve accuracy. Therefore, identification, characterization and compensation of these error sources are necessary to improve machine tool accuracy cost-effectively.

EXPERIMENTAL PROCEDURES

A schematic of the Sabre 750 VMC layout and temperature sensor locations is shown in Figure 1. Three material temperature sensors were magnetically attached to axis guides of X-axis (left side), Y-axis (back) and motor housing of Z-axis respectively. An air temperature sensor that was magnetically attached to machine table along with air pressure and humidity sensors were used to monitor environmental effects. Air pressure and humidity sensors were located inside the Environmental Controller unit (EC10). A Renishaw® laser measurement system is used during all linear displacement and angular error measurements. Schematic of laser interferometer set-up for linear displacement and angular error measurements are given in Figures 2 and 3 respectively. To obtain proper results, laser interferometer readings were corrected for changes in air temperature, humidity and pressure that affect refractive index of the air and the wavelength of the laser beam.

Prior to linear and angular error measurements using laser interferometer system, VMC was programmed to move in 3-axes with a programmed feed rate of 200 ipm and spindle speed of 3200 rpm to identify best temperature sensor placement locations. Then, linear/angular errors of SABRE 750 Vertical Machining Center were measured when the machine is in cold state. The basis of the measurement technique is to combine two coherent light (laser) waveforms. The two coherent waveforms are obtained by splitting the output beam from a single stabilized laser. These two beams (reference and measurement) are reflected back through the beamsplitter to form interference fringes at the detector (ML10 laser head). To simulate actual cutting process, the CNC machine was programmed to move in 3-axes simultaneously with a programmed feed rate of 300 ipm without spindle rotation for about 1-hour period. During warm-up cycle, three material sensors measured temperature distribution of the machine tool structure. After each warm-up cycle, machine was programmed to move in equal intervals in designated axis direction and dwelled for 4 seconds to measure machine's linear displacement/angular errors. Warm-up and error measurement test cycles were repeated for 8-hour period.

EXPERIMENTAL RESULTS

To identify most sensitive temperature sensor locations, the VMC was programmed to move in 3-axes simultaneously with a feed rate of 200 ipm and spindle speed of 3200 rpm. The more the temperature sensors are closer to the axis drive motor, the higher the temperature readings. Therefore, axis drive motors are identified as major heat sources for the VMC.

The X-axis linear displacement error is given in Figure 4. In the legend of the error graphs, Error (0-f) and Error (0-r) indicates that errors were measured when the machine was in cold condition (0 operational hour) for forward (f) and reverse (r) directions respectively. The error plots show that positional errors are generally lower at the beginning of axis travel (home position) and have linearly increasing trend with increasing axis nominal

position. Slope of the error plots gradually increases as the machine warms up. When the machine is in cold state, maximum linear displacement error is in the order of magnitude of 0.001". After 1-hour warm-up period, error immediately raises to 0.005". Maximum backlash (reversal) error is about 0" when the machine is in cold state and increases slightly above 0.001" and remains constant throughout the test. Maximum x-axis linear displacement error was obtained after 7-hours test period and the error magnitude was about 0.007".

Figure 5 summarizes linear displacement accuracy (LDA) and maximum reversal (backlash) error (MRE) for all 3-axes tested. X-axis had worst accuracy and backlash error followed by z and y-axes.

X-axis pitch error is given in Figure 6. Legend of the graph indicates that Pitch Error 0-f and 0-r measured when the machine was in cold state for forward and reverse directions respectively. The plots show linearly increasing trend with increasing X-axis position for both forward and reverse direction respectively. Generally X-axis pitch error is lower at the beginning of axis travel. The slope of the error plots gradually decreases as the machine warms-up. Therefore the colder the machine structure the higher the pitch error of the X-axis.

CONCLUSIONS

- Linear positional accuracy is best when the machine is in cold state and deteriorates with increasing machine operation (warm-up) time for all 3 axes.
- Generally all linear positional errors vary linearly with respect to axis nominal position and are highest at the end of axis travel range. Therefore machine home position gave best linear accuracy readings for X and Y-axes although Z-axis had worst accuracy at home position.
- X-axis, which has longest axis travel range, had worst accuracy and backlash error among the 3 axes being tested. Y-axis accuracy was better than Z-axis during first three-hour period and became slightly worse than Z-axis accuracy after 4-hour machine operation time.
- Z-axis motor temperature sensor gave the highest increase in temperature readings followed by X and Y-axes guide temperature sensors respectively. Axis drive motors were identified as major heat sources.
- X-axis pitch and Y-axis yaw errors were largest among all-angular errors of the machine.
- X-axis pitch / yaw errors and Y-axis yaw error linearly changed with axis nominal position. Y-axis pitch and Z-axis pitch /yaw errors showed non-linear trends with respect to axis nominal position.
- Generally all angular errors are higher towards end of the travel of the respective axis.
- Z-axis drive motor gave the highest increase in temperature readings followed by X and Y-axes guide temperature sensors respectively.
- During war-up and laser measurement cycles, no spindle rotation was allowed resulting in overall 4 °C decrease in Z-axis drive motor temperature. X and Y-axes readings remained unchanged.

ACKNOWLEDGEMENT

The financial support for the acquisition of the Cincinnati Milacron Vertical Machining Center used for this research by the National Science Foundation under grant #DUE9552065 with matching from the University of Missouri-Rolla are gratefully acknowledged. The Graduate Research Assistantship by the Intelligent System Center and Mechanical Engineering Department are also gratefully acknowledged.

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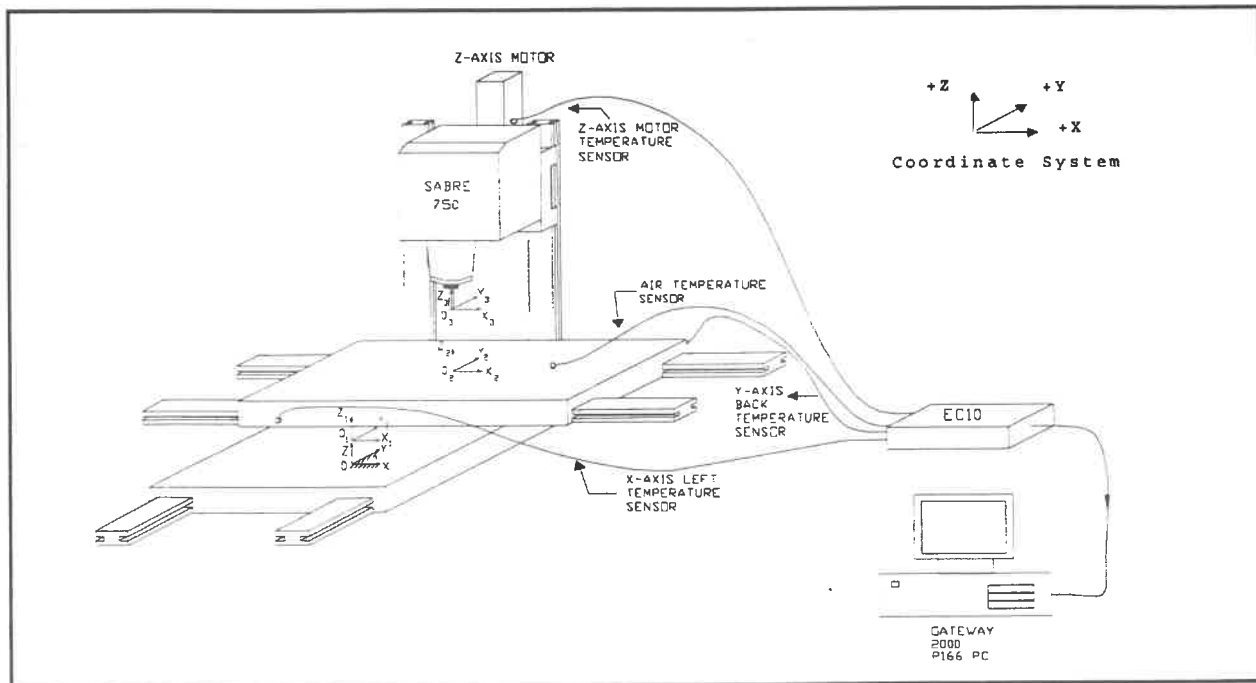


Figure 1. Schematic of the Sabre 750 VMC Layout and Temperature Sensor locations.

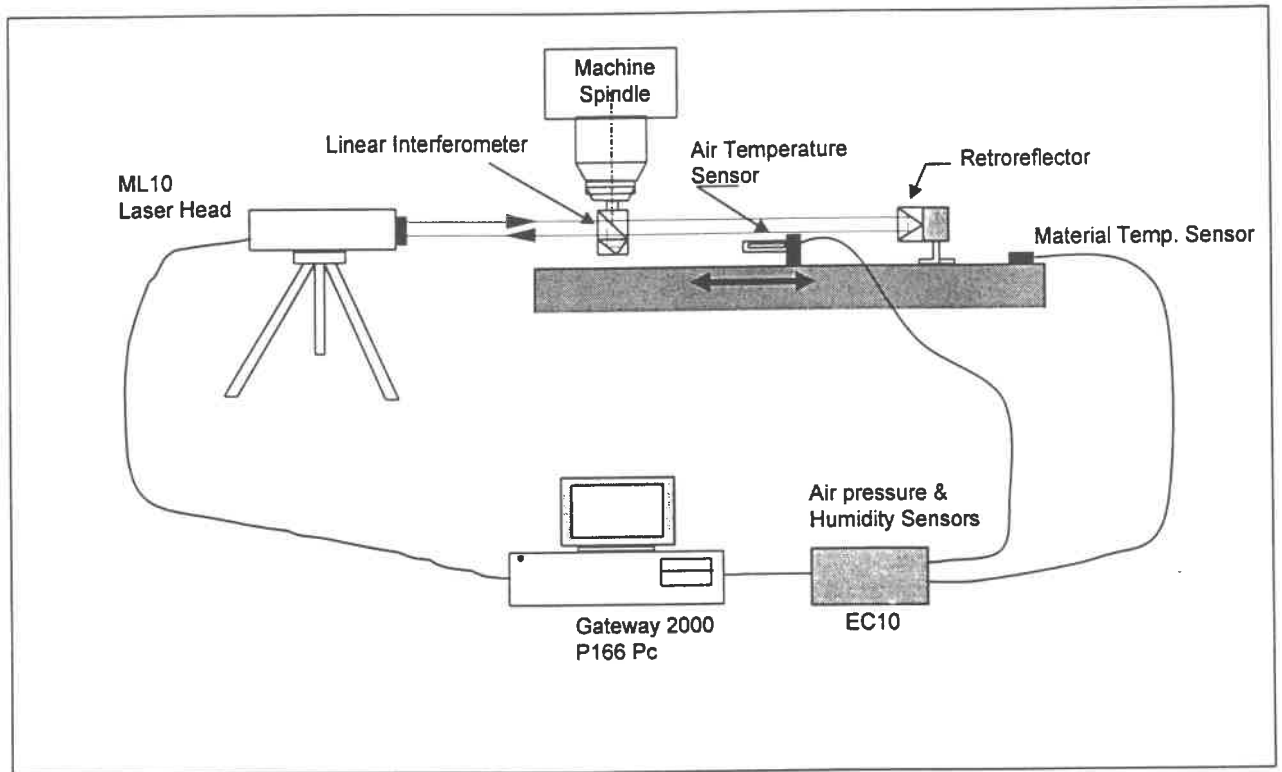


Figure 2. Schematic of Laser Interferometer set-up for linear displacement error measurement.

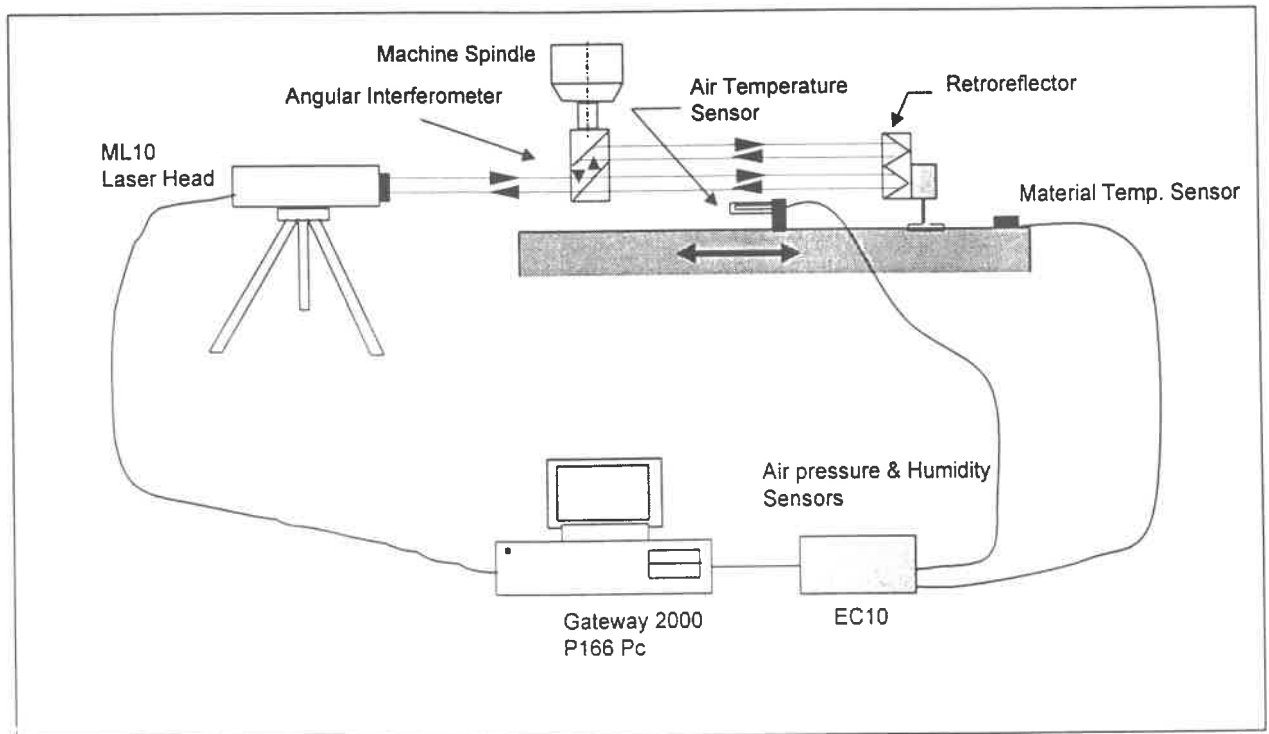


Figure 3. Schematic of Laser Interferometer set-up for angular error measurement.

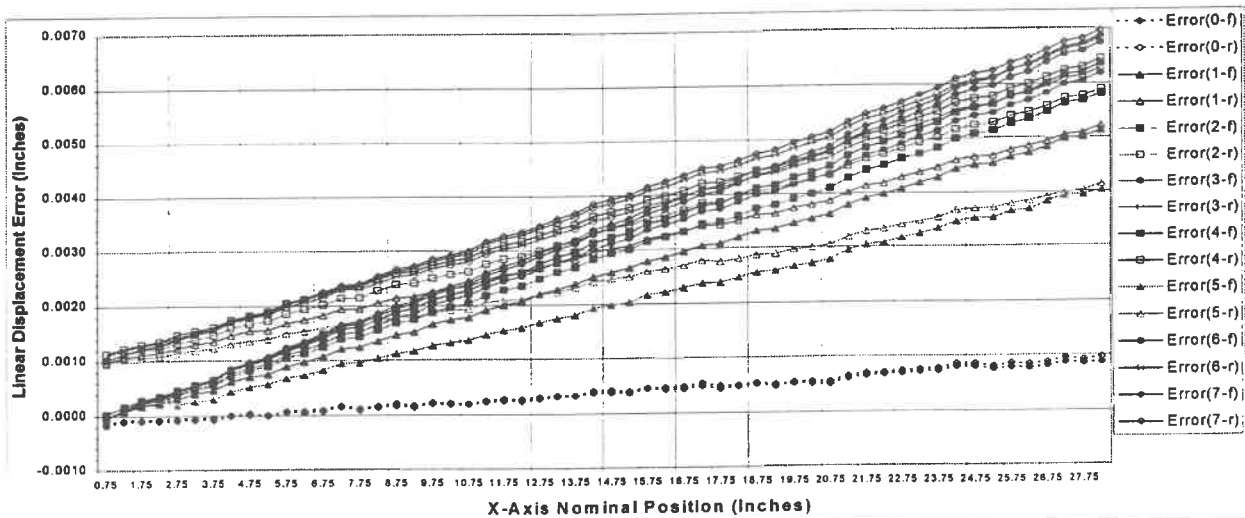


Figure 4. X-axis linear displacement error.

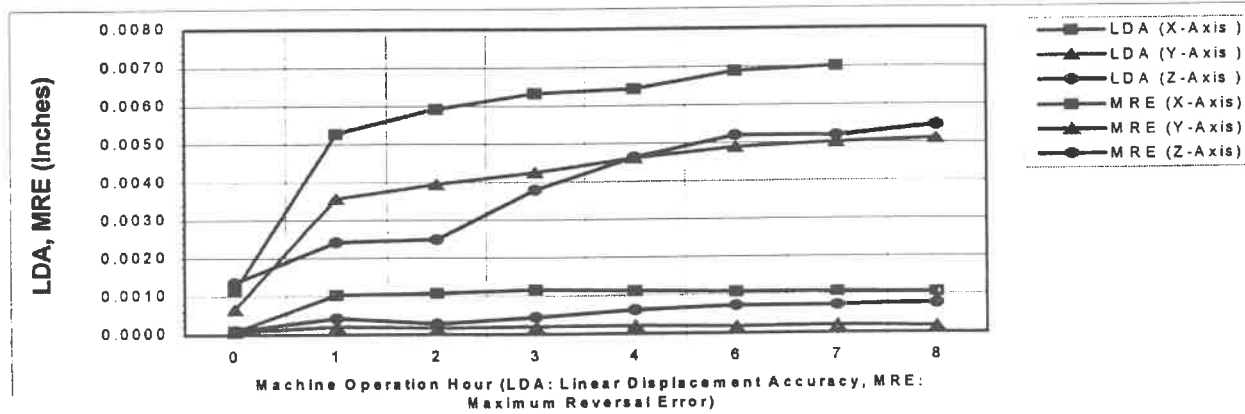


Figure 5. Linear Displacement Accuracy and Maximum Reversal Error (ASME B5.54).

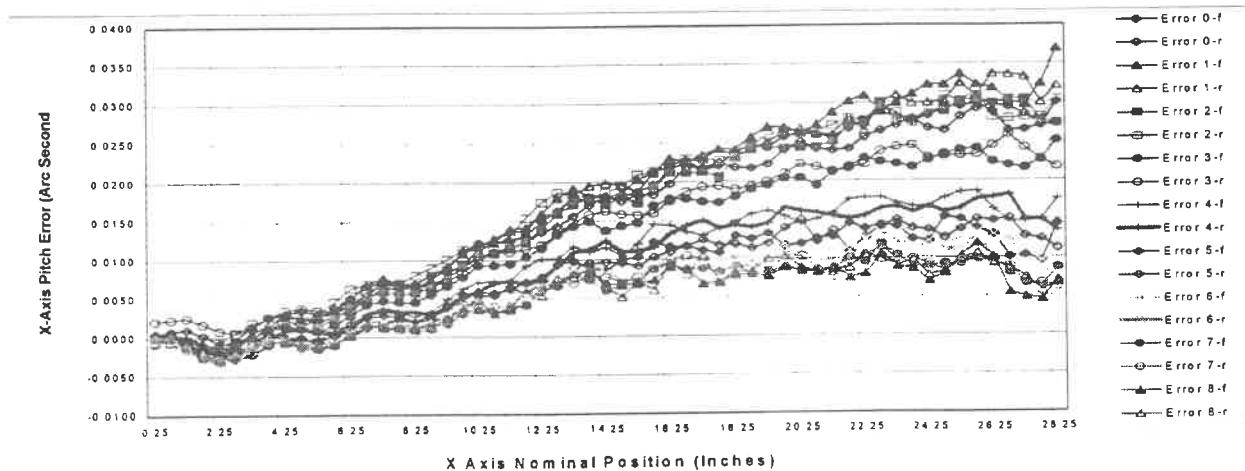


Figure 6. X-Axis pitch error.

ERRORS IN PROBE OFFSET VECTORS OF MULTIPLE PROBE ORIENTATIONS IN CMM MEASUREMENTS

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Introduction

Coordinate measuring machines (CMM) are widely used to inspect the dimensional quality of parts in modern manufacturing facilities. A typical CMM uses highly precise linear transducers, a microcomputer-based controller, a probe head, and a probe system. A probe head allows indexing of the probe system to discrete orientations in 3D as shown in Figure 1. The kinematic seat touch trigger probe is a commonly used probe type on modern CMMs and it consists of three parts: a stylus ball, a stylus shaft, and a probe body. Although the measuring or triggering force is small, a small elastic deflection occurs when the triggering signal is generated. As a result, the probe moves a small distance after the contact point and the distance between the contact point and the triggering point is known as probe pretravel or lobing. Pretravel is direction dependent and highly repeatable [1,2]. Figure 2 shows pretravel errors around equator of the test ball and around the 45° latitude of the test ball when a vertically oriented kinematic seat touch trigger probe with a 50 mm stylus extension is used.

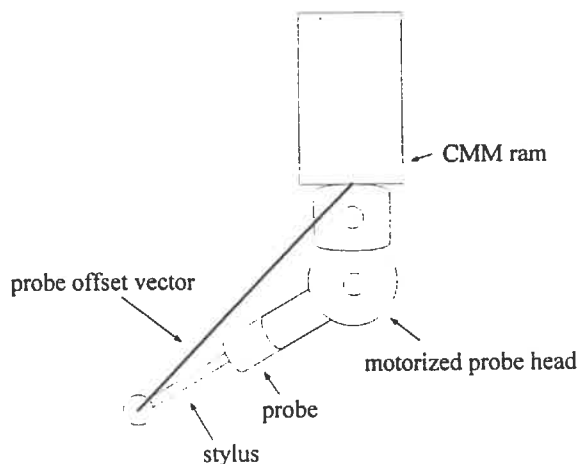


Figure 1 Illustration of probe offset vector

The probe pretravel is a major error source, and it has significant impact on the probe calibration process. The probe calibration is a process establishing the effective working diameter of the stylus tip and the probe offset vector. It is performed by the following steps: collecting a few measurement points on the test ball, fitting data points to a sphere, and calculating of the distance between centers of a test ball and a stylus tip. Current practice of probe calibration assumes that the probe pretravel is constant without observing the three-lobing patterns due to the direction dependent triggering mechanism of the kinematic seat touch trigger probes.

The probe-offset vector is defined as the location of the stylus ball center relative to the end of the CMM ram [3]. The stylus tip center is determined in the probe calibration process, and the probe offset vector can be obtained by calculating the distance between the center of the stylus tip and the end of the CMM ram. If the probe offset vector can not be correctly identified for each probe orientation indexed by the probe head, the measurement data from such multiple probe orientation setup would contain errors.

In this paper, we investigate the effect of probe pretravel on the calculation of probe offset vectors when kinematic seat touch trigger probes and indexable heads are used in CMM measurements. We compare the probe offset vectors calculated by using three methods: the conventional method, the contact force based method, and the SuperFit method.

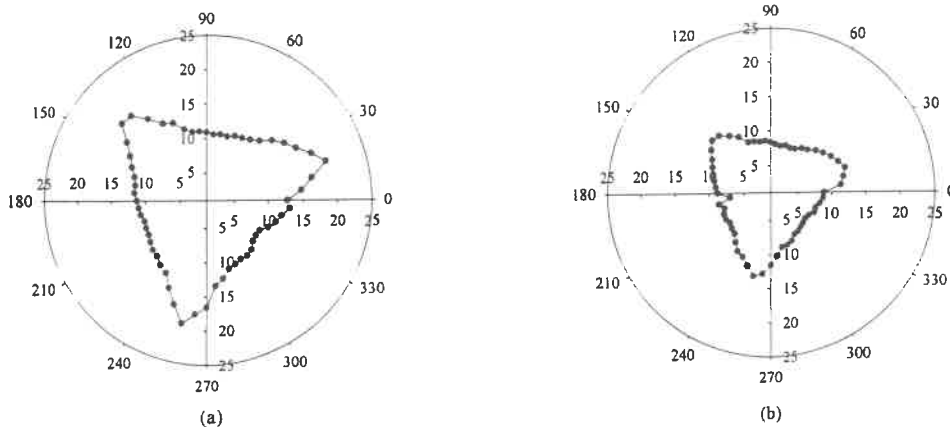


Figure 2 Probe pretravel errors around (a) equator and (b) 45° latitude of the test ball (50 mm stylus)

Calculation of probe offset vector

We introduce the three methods which can be used to calculate the probe offset vectors in multiple probe orientations (indexed by probe heads) when kinematic seat touch trigger probe is used.

Conventional method

The conventional method for establishing the probe offset vectors fit the probing data around a test ball to a sphere using the least squares method. The center of the test ball can be calculated and used as the basis for the probe offset vector for the specific probe orientation or position. Since the probe pretravel errors are systematic, not random, in nature, the fitted test ball center may not be close to the true center.

Contact force based method

The contact force based model is based on the derivation of the trigger force for kinematic seat touch trigger probes in the probing process [4,5]. The trigger force (F_p) is modeled as a function of the following parameters: F_s : spring force setting, F_t : threshold force level, W : weight of stylus structure, ϕ_0 : tripod phase angle, α : kinematic support angle, T : tripod leg size, L : stylus extension length, θ_w : probe orientation, l_w : center of gravity of the stylus structure, ϕ and θ are azimuthal and polar angles representing the probe approach direction. By treating the probe stylus as a cantilevered beam, the pretravel can be represented as:

$$\delta_{pt} = \frac{F_p (\sin \theta)^2 L^3}{3EI} \quad (1)$$

where δ_{pt} : model predicted pretravel. E : modulus of elasticity of the stylus shaft material, and I : moment of inertia of the stylus shaft. Define an objective function as

$$F_i = R - \left[\sqrt{(x_i - x_0)^2 + (y_i - y_0)^2 + (z_i - z_0)^2} + \delta_{pt} \right] \quad (2)$$

where R is the test ball radius. (x_i, y_i, z_i) is the i th measured X-Y-Z coordinate point. (x_0, y_0, z_0) is the calculated center of the test ball. and δ_{pt} is the pretravel at the i th measurement location. The value of F_i should close to zero if the test ball center location and the pretravel estimations are good. That is, the distance between the measured point and the estimated test ball center plus the estimated pretravel distance at this location should be very close to the test ball size. We formulate the least squares fitting of m measurement points, and the goal is to minimize the following objective function

$$S(Q) = \sum_{i=1}^m [F_i(Q)]^2 \quad (3)$$

where $Q = Q[R, x_0, y_0, z_0, \delta_{pt}(\phi_0, F_s, l, W)]$. We use modified Levenberg-Marquardt algorithm, which typically converges quickly and minimizes numerical problem [6], and IMSL Fortran math routines. Since the measurement data are defined in the machine coordinate system and the pretravel calculations are performed in a probe coordinate system, a coordinate transformation between the two coordinate systems is necessary when implementing the contact force based method with non-vertical probe orientations.

SuperFit method

The SuperFit method is the implementation of a mechanical model, for pretravel of kinematic seat touch trigger probes, developed by Estler et al. at the National Institute of Standards and Technology (NIST) [7,8]. The model accounts for the combined effects of rotational displacement, bending displacement, and frictional effects. The friction cap is defined by observing nearly constant pretravel behaviors due to the very small lobing amplitude for small polar angles. The rotational displacement is the rotation of the kinematic seat construction to generate trigger signal and is independent of stylus length. It has been shown that the SuperFit method produced an excellent estimate of stylus ball radius and minimized the probe-offset vector errors by accounting the probe lobing effects.

Evaluation of probe offset vectors using the three methods

Experiments were conducted on a computer-controlled CMM, and a TP2 5-way 3D touch trigger probe with a 50 mm straight stylus extension was used. In each experiment, 311 points of probe data, which represent 311 probe approach directions, were measured. These points were distributed in six (6) latitudes of the reference ball and one point at the North Pole. The polar angles of the seven latitudes are: 110°, 90°, 70°, ..., 30°, and 10°. Sixty (60) equally spaced points were measured in each latitude except 10 points on 10° latitude. Data were collected for various probe orientations indexable by a motorized probe head. In each probe orientation, ten sets of data have been collected. Three methods, including the conventional method, the contact force method, and the SuperFit method, are used to calculate the test ball center in various probe orientations.

Vertical case

The vertical case represents the most commonly used probe orientation in CMM measurements. The probe stylus is vertically oriented and pointing downward. The results for 10 sets of data calculated by the three methods are shown in Table 1. The contact force method and the SuperFit method give quite close test ball center locations. The difference in Z axis is around 0.35 μm , perhaps due to the difference in the models. While the conventional method gives a quite different test ball center in Z axis. There is a shift of around 6.0 μm in Z axis from the two model based methods.

Non-vertical cases

Tests have been done on various probe orientations ranging from 7.5° to 97.5°. However, we only show two orientations (45° and 90°) due to the space limitation. Table 2 shows the results from the 45° probe orientation and Table 3 shows the results from the 90° probe orientation. From the two model-based methods, the test ball centers are fairly close and the difference in each axis is less than 0.36 μm . While the conventional method gives a test ball center around 6.0 μm in distance; about 5.5 μm in Y axis and 3.5 μm in Z axis away from the two model-based methods. In horizontal probe orientation, X axis differences between two model based centers are sub-microns. Y- and Z- axis differences of two models are less than 0.33 μm . However, the centers of conventional method show about 5.3 μm and 1.7 μm in Y-axis and Z-axis respectively.

Acknowledgments

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Table 1 Calculated test ball center using three methods – vertical probe orientation

Conventional method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	0.76	0.40	0.77	0.69	0.84	0.54	0.57	0.60	0.72	0.66	0.66
Y [μm]	0.63	0.15	0.15	0.39	0.45	0.33	0.21	0.66	0.27	0.33	0.36
Z [μm]	-4.37	-4.38	-3.99	-3.96	-4.50	-4.25	-4.27	-4.06	-4.24	-4.30	-4.23

Contact-force method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	0.76	0.40	0.77	0.69	0.85	0.54	0.57	0.60	0.72	0.66	0.66
Y [μm]	0.63	0.16	0.15	0.39	0.45	0.32	0.21	0.66	0.26	0.34	0.36
Z [μm]	-10.31	-9.92	-9.79	-9.64	-10.31	-10.11	-9.82	-9.95	-9.80	-10.22	-9.99

Superfit method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	0.76	0.40	0.77	0.69	0.85	0.54	0.57	0.60	0.72	0.67	0.66
Y [μm]	0.63	0.16	0.15	0.39	0.45	0.33	0.21	0.67	0.27	0.34	0.36
Z [μm]	-10.30	-9.59	-10.35	-10.16	-11.01	-10.51	-10.43	-10.04	-9.98	-10.70	-10.31

Table 2 Calculated test ball center using three methods – 45° probe orientation

Conventional method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	-0.52	-0.28	-0.37	-0.50	-0.17	-0.47	-0.34	-0.49	-0.64	-0.49	-0.43
Y [μm]	-3.28	-3.36	-3.36	-3.36	-3.26	-3.30	-3.57	-3.44	-3.28	-3.33	-3.35
Z [μm]	-2.52	-2.43	-2.85	-2.44	-2.55	-2.65	-2.37	-2.68	-2.77	-2.67	-2.59

Contact-force method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	-0.53	-0.62	-0.68	-0.70	-0.67	-1.13	-0.92	-0.96	-1.02	-0.82	-0.80
Y [μm]	-8.27	-8.40	-8.73	-8.69	-8.74	-8.46	-9.12	-8.61	-8.63	-8.62	-8.63
Z [μm]	-5.81	-6.38	-5.38	-5.37	-5.78	-5.66	-5.96	-5.81	-5.95	-5.93	-5.80

Superfit method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	-0.55	-0.31	-0.40	-0.53	-0.20	-0.50	-0.36	-0.52	-0.67	-0.51	-0.46
Y [μm]	-8.66	-8.62	-9.26	-8.81	-8.96	-9.24	-8.87	-9.01	-9.09	-8.64	-8.92
Z [μm]	-5.95	-5.76	-6.35	-6.00	-6.34	-6.49	-5.88	-6.20	-6.56	-6.11	-6.16

Table 3 Calculated test ball center using three methods – 90° probe orientation

Conventional method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	0.04	-0.41	-0.12	0.31	0.19	-0.13	-0.06	0.08	-0.10	-0.16	-0.04
Y [μm]	-4.40	-5.05	-4.69	-4.79	-4.62	-5.15	-5.36	-4.68	-5.27	-4.82	-4.88
Z [μm]	-0.42	-0.62	-0.73	-1.08	-1.21	-0.86	-0.91	-1.46	-1.77	-1.73	-1.08

Contact-force method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	0.03	-0.46	-0.13	0.40	0.16	-0.14	-0.11	0.07	-0.14	-0.16	-0.05
Y [μm]	-9.62	-10.28	-9.90	-9.95	-9.65	-10.18	-10.34	-9.82	-10.37	-10.12	-10.02
Z [μm]	1.31	0.62	0.89	1.12	0.11	0.59	0.39	0.36	-0.24	-0.36	0.48

Superfit method	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6	Set 7	Set 8	Set 9	Set 10	10 set average
X [μm]	-0.01	-0.48	-0.15	0.27	0.14	-0.17	-0.13	0.01	-0.16	-0.19	-0.09
Y [μm]	-10.21	-10.52	-10.45	-10.56	-9.46	-10.51	-10.34	-9.80	-10.61	-10.57	-10.30
Z [μm]	1.90	1.46	1.09	0.67	0.65	0.72	0.95	0.58	0.20	-0.12	0.81

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Short Term Repeatability of Measurements of a Glass Polygon

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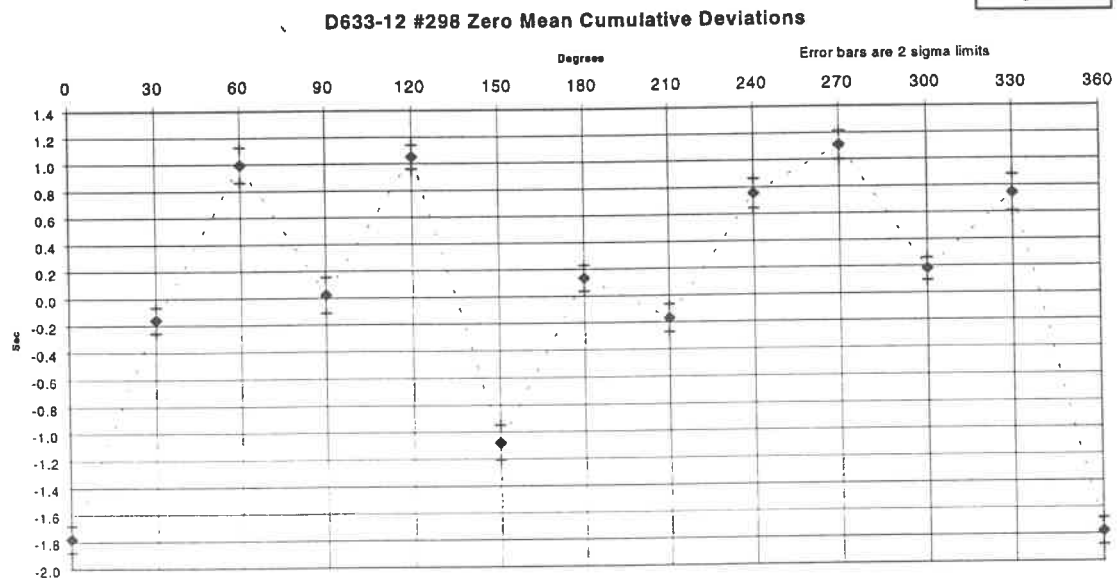
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Between the end of 1996 and mid 1997 the Air Force Primary Standards Laboratory tested a number of indexing tables that were purchased for use in the secondary metrology labs. The same polygon has been used during the testing of the index accuracy of the tables, affording a good opportunity to evaluate the short term repeatability of a glass polygon. The polygon used for these tests is a Davidson Optronics model D633-12, serial # 298. This is a solid glass polygon mounted in a steel housing. The polygon has a center bore to facilitate mounting on a spindle, and an annular base surrounding the center bore approximately 15mm wide with an outer diameter of about 40mm. The base is perpendicular to the mirror faces, within about 20 seconds. The autocollimator used for all the measurements was a Möller Wedel Elcomat HR, which has a readout resolution of 0.005 seconds, and a noise level of about 0.05 seconds in the lab environment.

In each case, the measurement was done with the "dual closure" method on 12 positions, the number of faces on the polygon. In a "dual closure" measurement each face of the polygon is compared to each corresponding position of the index table, so that a total of 144 angle differences are recorded (12^2 readings). In this measurement scheme, one obtains full information for both the indexing table and the polygon deviations.

Round robin tests of polygon measurements¹ have indicated that the "transferability" of polygon measurements from one lab to another is about ± 0.3 seconds. The data presented here, however, represent the "best case" repeatability for the polygon measurements. The same technician took all the data, and used the same autocollimator in each case. The index table was manually positioned between each measurement position but the autocollimator was read by computer after the manual move was completed. Figure 1 shows the average cumulative deviations for the

Figure 1



polygon over the 30 measurements done since mid December 1996.

In this graph, the data has been normalized so that the average of the data points in each individual measurement is zero before charting. This represents the average of all 30 different measurements of the polygon. The standard deviations range from about 0.04 seconds, to 0.07 seconds at any given position.

Looking at the difference from the average on each measurement results in Figure 2.

¹ "Program and Performance of International Comparison of Angle Standards" K. Toyoda and Y. Tanimura Bulletin of NRLM Vol. 37, No. 4 October 1988

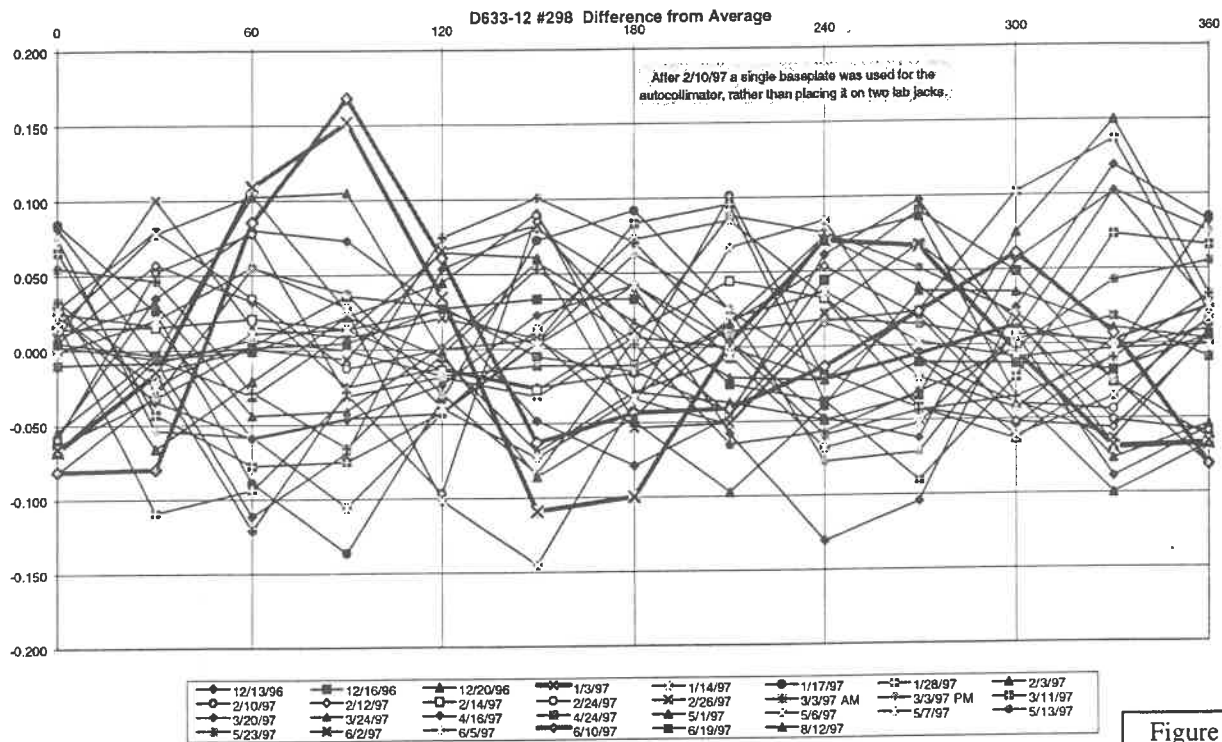


Figure 2

Although this graph is extremely crowded, there are a few things that can be noticed. First, one would expect that the difference from the average in any given measurement would be randomly distributed across by position. Instead, a generally sinusoidal pattern of the deviations with 2 peaks per circle can be seen in each case. Secondly, some of the measurements seem to repeat, in several cases even though they are well separated in time from one measurement to the other. Notice particularly the measurements of 1/3/97 and 6/10/97 (Highlighted). Both have a similar pattern for the location and magnitude of their peaks and valleys, even though the measurements occurred more than five months apart. Since each measurement used a different index table, the setup was completely rebuilt between each measurement. Lastly, all the measurements, except for a couple of "outliers," fall within ± 0.15 seconds of the average.

The sinusoidal pattern of the deviations can perhaps be understood as the result of small centering errors of the polygon alignment to the rotation axis of the indexing table. Such a deviation would cause the position of the polygon faces to translate across the width of the autocollimator beam as the table rotates. The alignment of the polygon is such that centering deviations greater than 1 mm are unlikely. The polygon mounting spindle is set concentric to the indexing table rotary axis within 0.12 mm, and the clearance between the spindle and the center bore of the polygon is about 0.25 mm maximum. Work at NIST² has indicated that small amounts of decentering can cause deviations in the 0.1 to 0.2 second range, although the decentering in the NIST study was larger, about 5 mm. If this is the source of the observed sinusoidal pattern, it could also explain the fact that several of the measurements repeat each other better than the overall repeatability of the measurement. Perhaps occasionally the specific alignment of the polygon's position repeats a previous setup, causing an approximate duplication of the measurement. If the measurement is simply run twice, with no change in the setup, the repeatability between the two is about 0.03 sec. From a practical standpoint of verifying polygon measurements, however, this repeatability figure is meaningless, since it primarily is a measure of autocollimator repeatability.

The deviations from the average are plotted below as a 3D surface (Figure 3). Notice that the graph becomes less "mountainous" for about 4 measurements after 2/12/97. There was a change in the measurement set up that occurred for the measurements at this time. Prior to this, the autocollimator was supported by two lab jacks, one beneath the two fixed front feet, and the other beneath the rear adjustable foot. After 2/12/97, the autocollimator

² "Advanced Angle Metrology at the National Institute of Standards and Technology." W. T. Estler, Y. H. Queen, D. Gilsinn and D. Pieczulewski. *Proceedings of the American Society for Precision Engineering*, vol. 4 1991

was set on top of a single aluminum baseplate, which was supported by the two lab jacks. For a time, this seemed to reduce the amount of “noise” in the measurements.

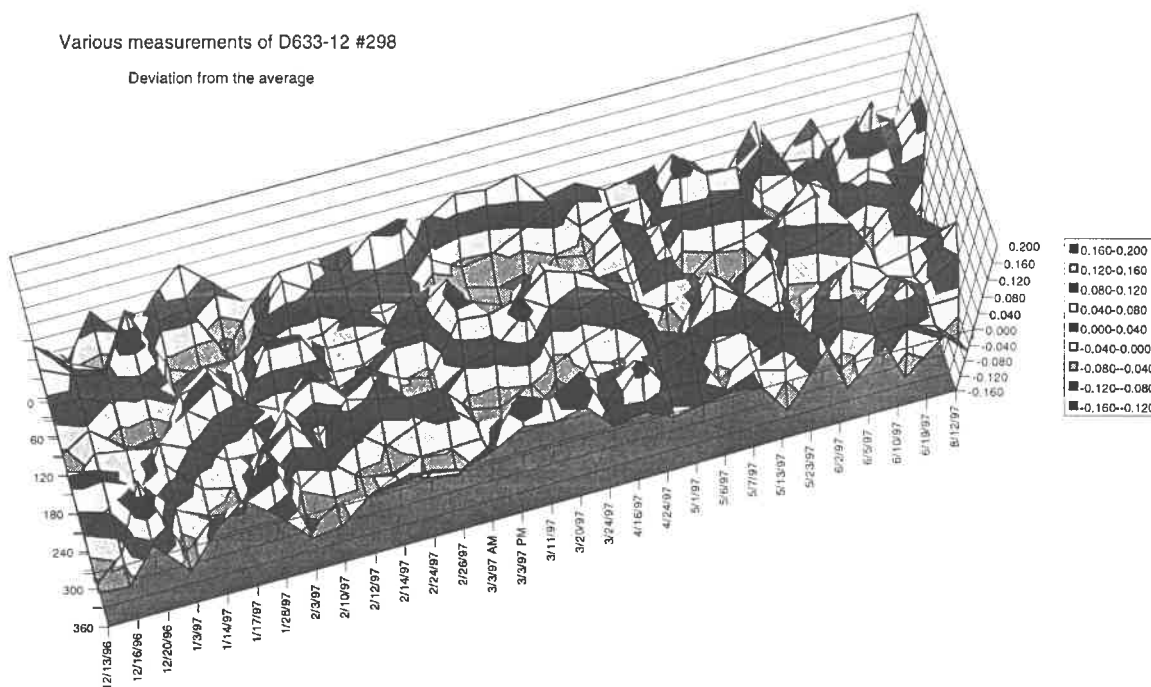
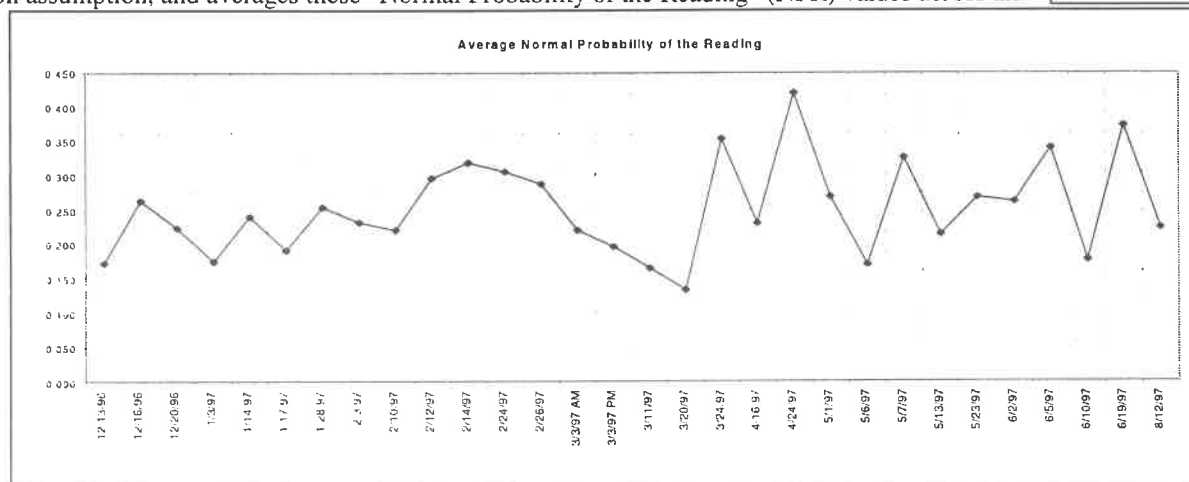


Figure 3

A method of quantifying the amount of noise in the measurement was desired. If one makes the assumption that the measured deviations at any given polygon face are normally distributed across the measurements done over time, a standard deviation of the calculated deviations can be computed for each position. If one then calculates the probability that a given deviation from the average would be seen based on the normal distribution assumption, and averages these “Normal Probability of the Reading” (NPR) values across all

Figure 4



positions of the polygon, for a given polygon measurement, one gets a number that can be taken as a “figure of merit” for the measurement. In these calculations, the absolute value of the deviation from the average was used, so all of the NPR readings will be less than or equal to 0.5. Figure 4 is a chart of the average of these values for each individual measurement of the polygon.

The average NPR seemed to increase slightly when the aluminum base plate was put into use. However, the values then started to decrease in a systematic manner after a couple of weeks. One would expect that the average

NPRs would be randomly distributed, a systematic pattern in these values is worrisome. One hypothesis that might explain this sort of pattern is that the polygon itself was changing. Referring back to Figure 3, one can see that after approximately 2/24/97 a pattern is beginning to emerge. While this part of the chart is getting more "mountainous," it is also developing some fairly well defined "channels" and "ridges" along the time axis of the plot. This could be consistent with a change to the polygon itself. If this was happening, then one could imagine a gradual decrease in the average NPRs as the new values were compared against the old average. After 3/20/97, however, the average NPRs changed to what appears to be a more random behavior again. At the end of Figure 3 one can see as well that it appears that the "channels" and "ridges" have disappeared, and the more "mountainous" appearance of the first part of the graph has returned. There was no known change in the measurement technique at this time. Lab temperature was controlled to within $\pm 0.6^\circ\text{C}$ during the course of these tests. It is not generally believed that temperature variations of this order will cause changes in a glass polygon of this magnitude, but specific studies to test this have not been done, to the best knowledge of the author.

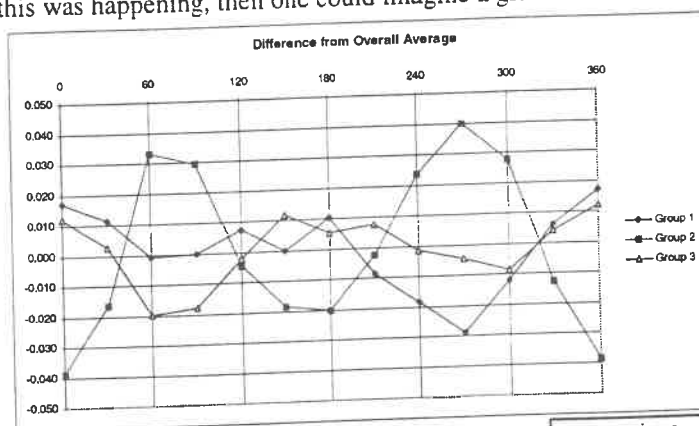
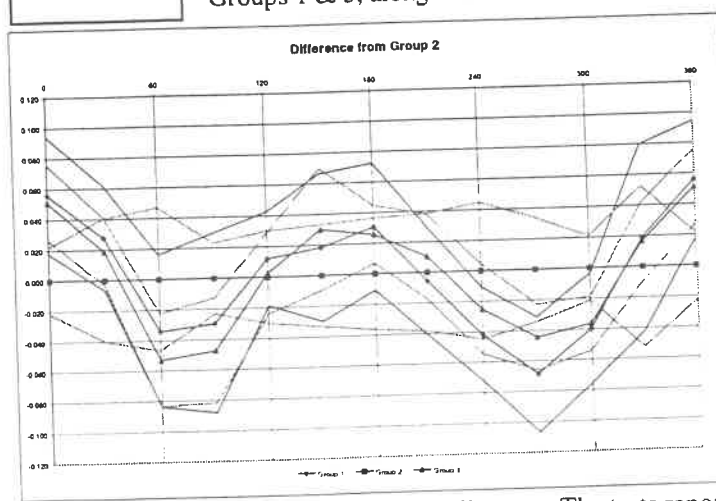


Figure 5

It appears, from Figure 4, that there are three distinct groups of data here. The first group consists of the measurements done before 2/12/97. Group 2 consists of the measurements done from 2/12/97 to 3/20/97, and Group 3 has the measurements done after 3/20/97. Figure 5 shows the difference of the average of each of these groups to the overall average of all the measurements. It seems obvious that the second group, those measurements taken between 2/12/97 and 3/20/97, are from a different population than either the measurements taken either before or after this period, and that Groups 1 and 3 likely belong to the same population. However, taking the uncertainty of the measurement into account, this is less clear. Figure 6 shows the difference from Group 2 for Groups 1 & 3, along with the uncertainty bands for each group. These differences are all entirely

Figure 6.



within 2σ of the mean ($2\sigma/\sqrt{n}$) with the single exception of the 0° face. In this case the Group 2 and Group 3 uncertainty bands fail to overlap by only 0.005 seconds. In one other case, the one at 300° , the uncertainty bands of Group 2 and Group 3 overlap by only 0.003 seconds.

Conclusions

The "best case" repeatability of the measurement of a polygon is approximately ± 0.15 seconds of arc, at a coverage factor of 2. This is not the same as the "transferability" of the polygon measurement from one user to another, since in this case, the same technician did all the

tests, using the same autocollimator in all cases. The tests reported upon here do not consider the variations caused by differences between different measuring instruments, and minor differences in technique between one user and another. While it would be desirable to eliminate these effects, this is not possible with the current state of the art.

It is unclear whether or not small changes over time have been detected in the measurement of this polygon. These data present a tantalizing hint of a difference, but the size of the differences are too small to make any firm conclusions that there was, in fact, a change between the three groups. The correlations look good in Figures 5 & 6 above, but they are all essentially buried within the measurement noise.

Acknowledgements: I'd like to thank David Brooks, of Wyle Laboratories, whose work in taking these data made this report possible.

CHARACTERIZATION OF HYDRAULIC PISTONS WITH FORM ERROR BY CONFOCAL MICROSCOPE

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INTRODUCTION

The functional behavior between piston and cylinder under hydrodynamic lubrication (viscous friction zone) in a hydraulic actuator is studied by considering diverging tapered pistons having different types of form errors like elliptic, 3-lobe and 5-lobes [1]. The characteristic equations were derived using Reynolds equation considering the film thickness variation in the clearance between piston and cylinder taking into account only the out-of-roundness and taper of the piston. The out-of roundness parameter at a particular cross section of the piston may not give a correct evaluation of the frictional aspects in a hydraulic actuator. The 3-D surface topography parameters have to be considered for functional evaluation of the surface as the surface interacts 3-dimensionally. In the present investigation, the characterization of the profiled pistons has been carried out by measuring the 3-D surface topographic errors (micro and macro errors) using confocal microscope based on focus detection technique [2]. The theoretical estimates of the friction parameter is obtained using dimensional analysis by a set of functional parameters calculated from the 3-D measurement of the pistons using confocal microscope along with the operational parameters of the hydraulic actuator. In order to ascertain the salient 3-D topography parameters as applicable to the present investigation on friction between piston and cylinder, the bearing area curves of the test pistons are analyzed.

MEASUREMENT AND CHARACTERIZATION

The pistons with elliptic and 3-lobe profiles with different l/d ratios were fabricated using a horizontal machining center. The cutter path was programmed depending upon the profile. The surface irregularities like roughness, roundness and cylindricity of these pistons have been measured in the present investigation using the confocal microscope. The measurement principle and schematic diagram of the confocal microscope is already given [2]. The 2-D and 3-D surface topography of 3-lobe and elliptic pistons are shown in Fig. 1. Both axial and circumferential tracings are taken to study roundness and roughness characteristics of cylindrical pistons. The measurements have been carried out for a length of 3.2 mm along the axis with a step interval of 100 μm and 400 data points around the circumference. The mean diameter of the pistons is 20 mm. From the measured data, the surface parameters are calculated using least squares mean plane as the datum with out filtering the measured surface. The software then draws the polar representations of roundness profiles and 3-D plots of the pistons. Some of the calculated 3-D surface topography parameters of the 3-lobe, elliptic and circular pistons are given in Table 1. Both statistical (S_a , S_q , S_{sk} and S_{ku} etc.) and functional parameters (S_{pk} , S_k and S_{vk}) are calculated to characterize both the general and the stratified topography.

Table 1: Surface Topography Parameters of Various Pistons

Piston Nos Parameters	13-3L	12-3L	9-3L	21-E	19-E	31-E	Circular
$sP_a, \mu m$	1.83	2.39	2.028	1.59	2.78	1.84	0.301
$sP_q, \mu m$	2.24	2.77	2.387	1.89	3.10	2.18	0.539
$sP_z, \mu m$	10.22	10.61	10.02	8.21	11.92	9.29	4.115
$sP_t, \mu m$	13.739	13.34	12.96	12.85	12.79	12.62	5.734
sP_{sk}	-1.007	-1.016	0.057	-1.465	-0.129	-0.062	2.461
sP_{ku}	3.143	2.110	2.264	2.47	1.811	2.175	13.268
$sP_{ds} 1/mm^2$	18.39	27.80	27.45	28.66	19.86	28.66	17.11
$sP_{\Delta q} 1/\mu m$	0.0191	0.0103	0.0112	0.0118	0.0157	0.013	0.0039
$sP_{sc} 1/\mu m$	0.073	0.034	0.029	0.036	0.038	0.035	0.012
$sP_{dr} \%$	0.423	0.4230	0.423	0.423	0.632	0.507	0.433
$sP_{pk} \mu m$	2.46	2.96	1.878	1.59	1.83	1.837	1.564
$sP_k \mu m$	6.23	6.13	5.23	7.14	7.86	7.14	1.272
$sP_{vk} \mu m$	4.12	3.91	4.29	2.52	3.36	3.131	2.212

Therefore the surface functional parameters obtained by bearing area ratio curve, the taper of the piston and the operational parameters like side load, pressure and velocity are considered here as the parameters relating to viscous friction force (F_f) in a hydraulic actuator.

The parameters considered are :

w = Side load, N

P = Operating pressure, N/m^2

u = Piston velocity, m/s

l = Length of the piston, m

d = Nominal diameter of the piston (m) = $2r$

c = Radial clearance, m

η = Viscosity of the oil, Ns/m^2

t = Taper, m

sP_{pk} = Reduced surface peak height, m

sP_k = Surface core depth, m

sP_{vk} = Reduced surface valley depth, m

DIMENSIONAL ANALYSIS BY BUCKINGHAM π THEOREM

Dimensional analysis is considered here for developing an equation relating viscous friction force (F_f) and the influencing parameters listed above.

Let

$$F_f = f(w, u, c, p, r, l, t, \eta, sP_{pk}, sP_k, sP_{vk})$$

$$\text{Therefore } \phi(F_f, w, u, c, p, r, l, t, \eta, sP_{pk}, sP_k, sP_{vk}) = 0 \quad (1)$$

The above variables are grouped together suitably to form 7 quantities and the function contains 3 fundamental dimensions. This gives $n = 7$ and $m = 3$. Therefore according to Buckingham π theorem, there are $[(n-m) = 4]$ non-dimensional factors, which are the π terms.

$$\text{Therefore } \phi(\pi_1, \pi_2, \pi_3, \pi_4) = 0$$

The π terms in the above equation are obtained by grouping the variables given in Eqn. 1, we get the final equation as

$$\frac{\mu r}{c} = K \left(\frac{plc}{w} \right)^x \left(\frac{\eta uc}{w} \right)^y \left(\frac{sP_{pk} sP_k t}{sP_{vk} c^2} \right)^z \quad (2)$$

VISCOUS FRICTION FORCE CHARACTERIZATION

The constants K , X , Y and Z in Eqn. 2 are evaluated from experimental data of the friction force of four pistons chosen from the set of pistons given in Table 1. The various parameters of these four pistons as also the measured friction force at certain operating condition for each piston are given in Table 2.

Table 2 : Simulated Model and Friction Force

Sl. No.	Piston No.	$sP_{pk} \times 10^{-6}$ m	$sP_k \times 10^{-6}$ m	$sP_{vk} \times 10^{-6}$ m	$t \times 10^{-6}$ m	$l \times 10^{-3}$ m	$d \times 10^{-3}$ m	$c \times 10^{-6}$ m	w N	u m/s	p bar	F_f N
1	13-3L	2.46	6.23	4.12	6	40	20	15	1.7	0.15	6	1.10
2	12-3L	2.96	6.13	3.91	11	20	20	14	1.3	0.16	5	1.01
3	21-E	1.59	7.14	2.52	11	40	20	21	1.7	0.15	5	1.04
4	19-E	1.83	7.86	3.36	8	20	20	17	1.3	0.13	4	1.20
Oil viscosity = 0.065 Ns/m ²												

Substituting the results from Table 2 in Eqn. 2 and solving ,gives

$X = -0.5$, $Y = -1.25$, $Z = 0.25$, and $K = 5.513 \times 10^{-7}$. Hence the relation for the friction parameter is given by

$$\frac{\mu r}{c} = 5.513 \times 10^{-7} \left(\frac{w}{plc} \right)^{1/2} \left(\frac{w}{\eta uc} \right)^{5/4} \left(\frac{sP_{pk} sP_k t}{sP_{vk} c^2} \right)^{1/4} \quad (3)$$

The friction parameter can be calculated by using the above relation at different operating conditions for all piston velocities in the viscous operating zone.

BEARING AREA RATIO CURVES

Fig 2 shows the bearing area ratio curves of pistons 12-3L, 19-E and circular piston with an l/d ratio of 1 for all the pistons. Fig 3 shows the bearing area ratio curves of 13-3L and 21-E pistons with an $l/d = 2$. As seen from Table 2, the value of sP_{pk} and sP_{vk} are more for the 3-lobe piston compared to the elliptic piston. This indicates short running in and good lubrication property of the 3-lobe piston. The parameter sP_k of the 3-lobe piston is less compared to the elliptic piston and hence the load carrying capacity of the 3-lobe piston is more than the elliptic piston. Further from Fig 2 and Fig.3 it can be seen that the 3-lobed piston surface is the better one compared to the elliptical piston with respect to the specified functional requirement (friction between piston and cylinder) because it has a lower sP_k and a higher sP_{vk} (meaning higher load carrying capacity and high oil retaining capacity). Therefore viscous friction force of the 3-lobed piston is less compared to the elliptical piston. On the other hand, the circular piston, having a small roundness error of $4\text{ }\mu\text{m}$ carries more load than the 3-lobe and the elliptic pistons. This is due to the small value of sP_k compared to 3-lobe and elliptic pistons. For a piston without any roundness error or taper, which is not possible in actual practice, Eqn. 3 will not be valid.

RESULTS AND DISCUSSION

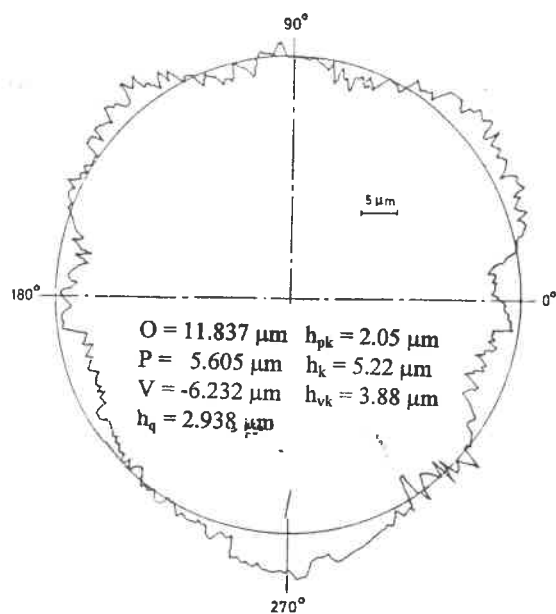
The friction parameter variation with piston velocity for different pistons including pistons with form error and approximately circular piston are indicated in Fig 4. In these figures the friction parameter variation with velocity obtained by Eqn. 3 has been incorporated by drawing a best fit line through the calculated points. The result indicate that there is a good correlation between the relationship obtained by dimensional analysis and the experimental results. Further, it is also observed that the friction parameter values under similar operating condition are generally lower for 3-lobe pistons as compared to elliptic pistons in line with the bearing area ratio curves. Also the approximately circular piston has the lowest value for the friction parameter. Thus the functional parameters (sP_{pk} , sP_k , sP_{vk}) obtained from the 3-D topographic measurements of the pistons with form error have an influence on the friction parameter.

CONCLUSION

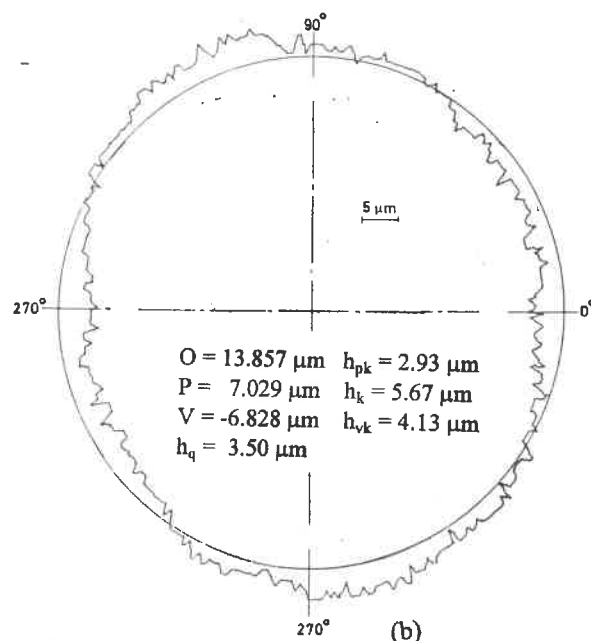
In the earlier works, the friction parameter was estimated by calculating the film thickness variation around the piston inside the cylinder bore by considering theoretically definable form deviations and solving the Reynolds equation. In order to facilitate better representation of the friction parameter variation with velocity of such pistons working in the viscous operating region, the 3D surface topographic errors were measured using confocal microscope. Based on the three-dimensional characterization of the pistons, the frictional aspects in a hydraulic actuator are modelled. Dimensional analysis has been carried out to find a relation between the friction parameter and the functional parameters of the measured surface. The cylindricity error, and the three functional parameters like sP_{pk} , sP_k and sP_{vk} along with operational parameters gives a better characterization of the frictional aspects in a hydraulic actuator than the conventional form error values like the out-of roundness at a particular section and the taper of the piston. The theoretical estimate of the friction parameter by the present analysis reduces the complexity involved in the calculation of the same using Reynolds equation.

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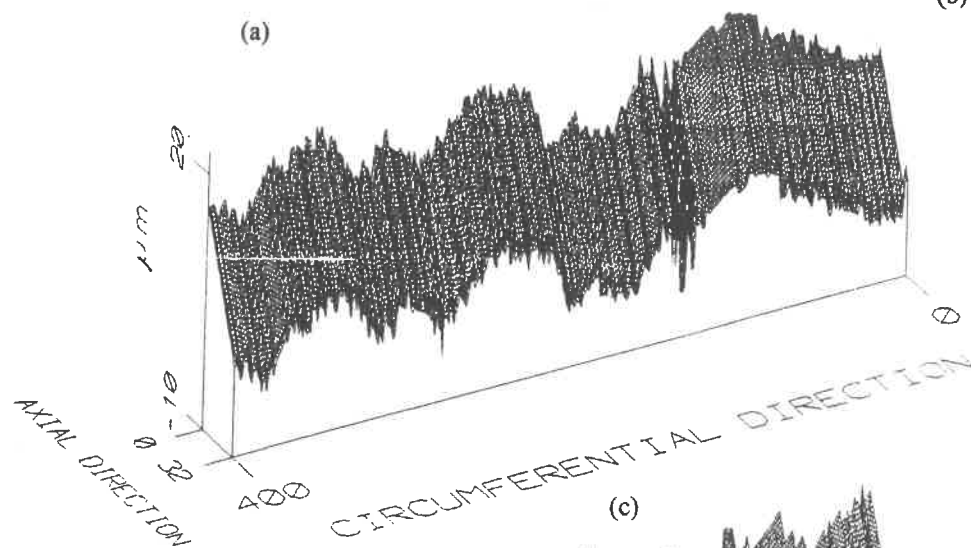
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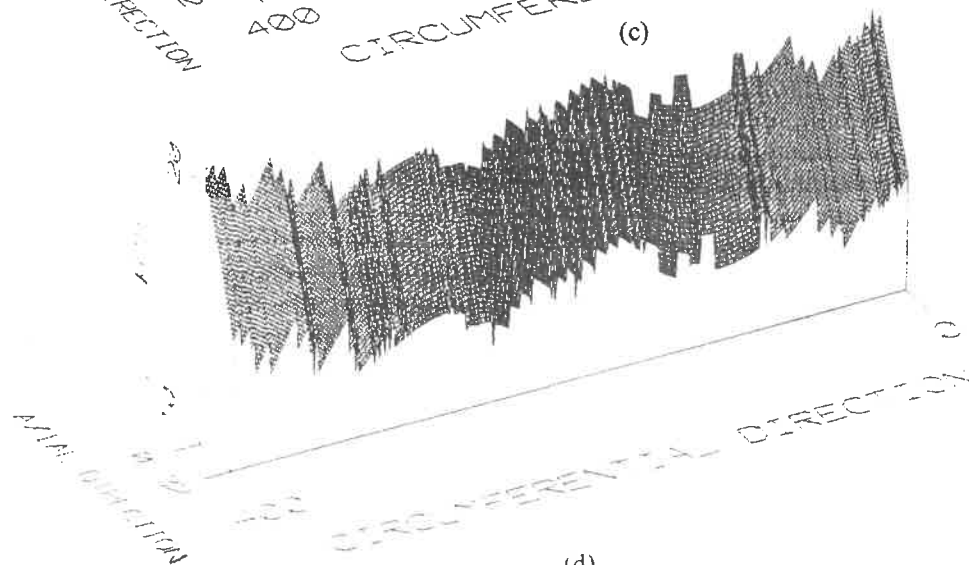
(a)



(b)



(c)



(d)

Fig. 1 The 2-D and 3-D surface topography of 3-lobe (a and c) and elliptic pistons (b and d).

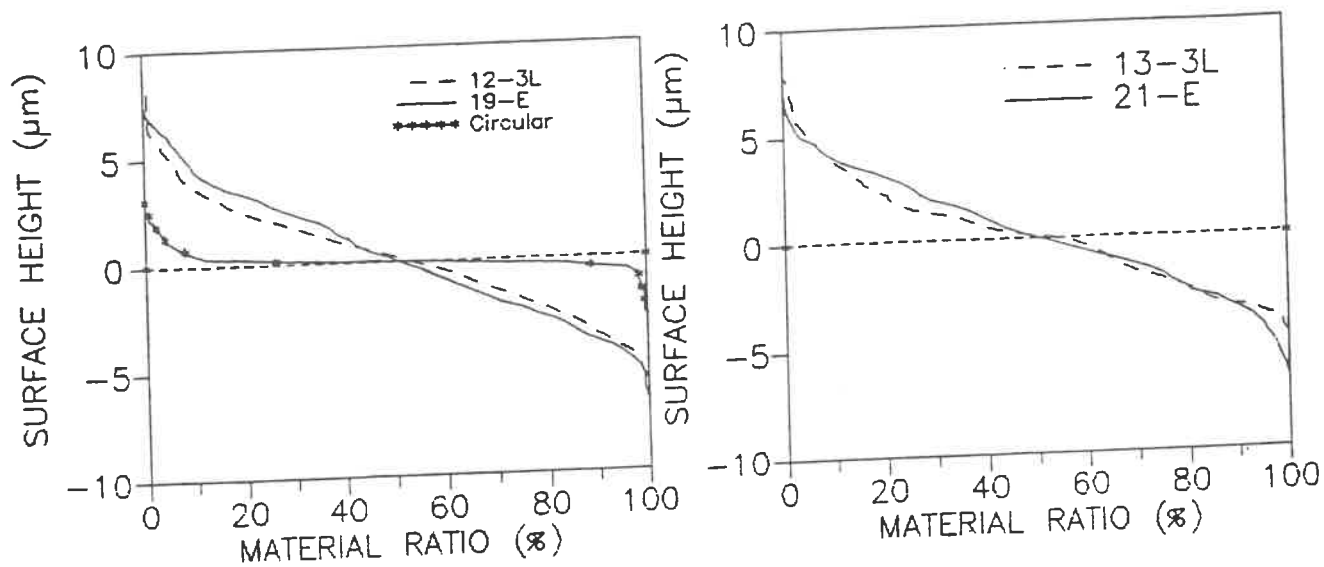


Fig. 3 Bearing Area Ratio Curves of 3-lobe (12-3L) Elliptic (19-E) and Circular (5-C) Pistons ($l/d = 1$)

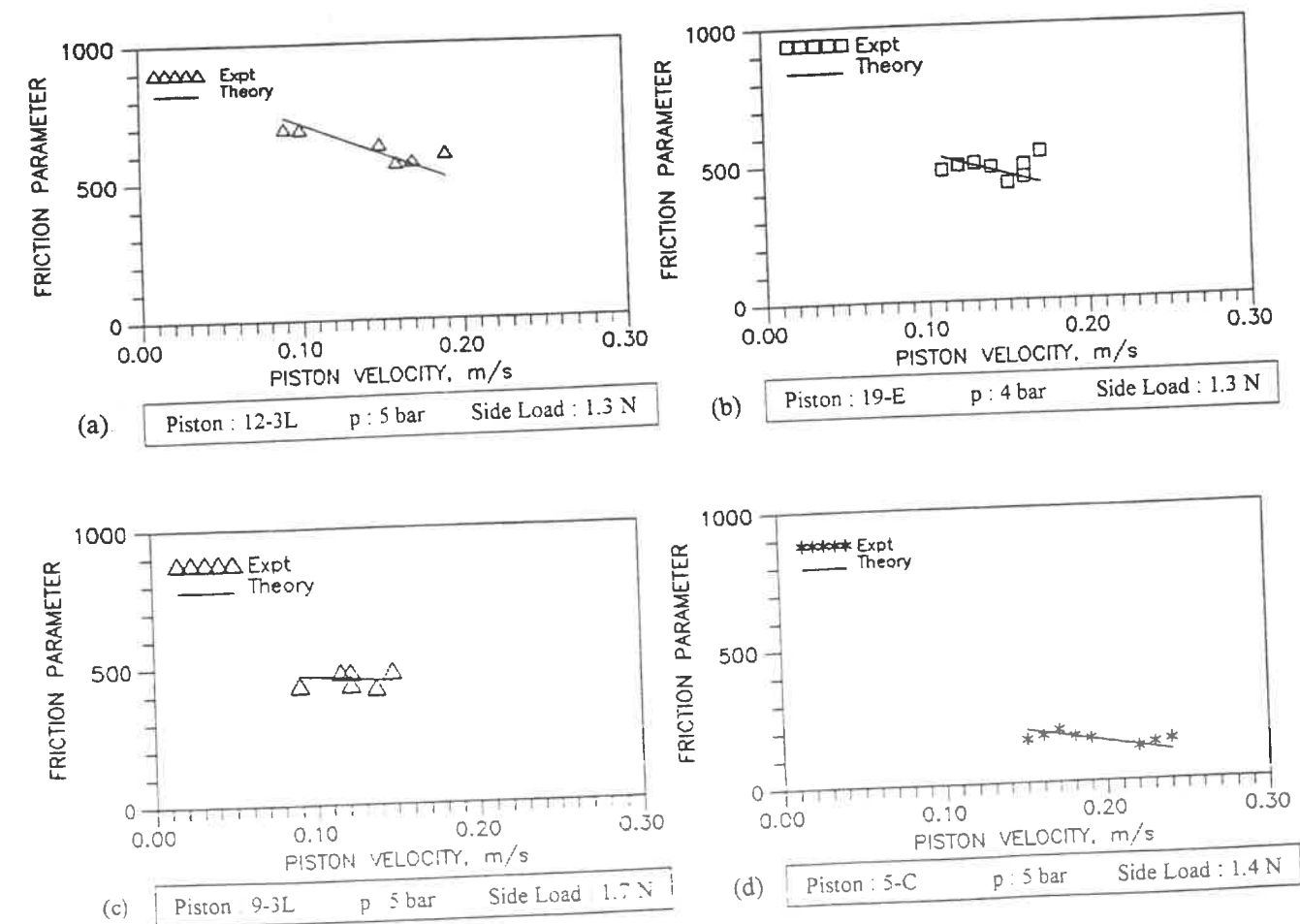


Fig. 4 Variation of Friction Parameter vs Piston Velocity for Various Pistons

Absolute Calibration of Nanoradian Angle Sensor

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ABSTRACT

The present study proposes a new method for calibrating the mean sensitivity of a precision angle sensor with an optional small measurement range. Using this method and the previously proposed in situ self-calibration method of the linearity error of an angle sensor, the output curve of the angle sensor can be absolutely calibrated. An angle sensor which has a measurement range of 500 microradians (μrad) was absolutely calibrated by the methods. The repeatability error for calibrating the mean sensitivity was approximately $\pm 0.01\%$ of the measurement range, and that for calibrating the linearity error was approximately 35 nanoradians (nrad).

1. INTRODUCTION

The authors proposed two self-calibration methods for angle sensor that do not require an additional reference instrument [1][2]. In one case, the in situ self-calibration method of a single angle sensor [1], and thus does not require an additional angle sensor as our other proposed method does [2]. However, this method is based on the premise that the mean sensitivity of the sensor (inclination of linear calibration line) is given beforehand, so that this method can only determine the linearity error, in other words, the deviation from the linear calibration line. Our other method have referred to as absolute calibration which can determine autonomously not only the linearity error but also the mean sensitivity [2]. However, it is difficult to calibrate the mean sensitivity of an angle sensor with a small measurement range using this method.

The present study proposes a new method for calibrating the mean sensitivity of an angle sensor with a small measurement range. Using this method and the in situ self-calibration method of the linearity error of an angle sensor [1], the output curve of the angle sensor can be absolutely calibrated.

2. CALIBRATION PRINCIPLE OF MEAN SENSITIVITY

Figure 1(a) shows a schematic diagram of the previously proposed self-calibration system to determine the mean sensitivity of an angle sensor [2]. The system in Fig. 1(a) consists of two coaxial rotary tables that face each other. These tables are driven separately by two stepping motors. An angle sensor with its light axis directed toward the center of the rotation is placed on rotary table 1, and a plane mirror is installed in the center of rotary table 2. The principle of the calibration is based on the fact that the accumulation of dividing errors of a rotational axis reduces to zero after every turn [3]. α in Fig. 1(a) is the measurement range of an angle sensor. However, the system requires approximately 2h to record a set of data for calibration even if the measurement range of the angle sensor is approximately 6 mrad [2]. If the calibration time is too long, the calibration accuracy is strongly influenced by the thermal drift of calibration system. As a result, this method is not suitable for calibrating an angle sensor with a measurement range less than 6 mrad.

Figure 1(b) shows the newly proposed method to determine the mean sensitivity of an angle sensor. This method can greatly shorten the calibration time of the previously proposed method, and can calibrate the mean sensitivity of an angle sensor with an optional measurement range. The calibration procedure involves repetition of the following steps during a single rotation of rotary table 1, such that

measurement interval β ($\beta \geq \alpha$) is $1/N$ of 2π :

- (1) The angle sensor is adjusted to the minimum value M_L of the angle sensor measurement range $T(\theta_{\min})$ of Fig. 1(b). Then M_{Li} is measured.
- (2) After the plane mirror on the rotary table 2 is rotated by α , the M_{Hi} of the maximum value M_H of the angle sensor measurement range $T(\theta_{\max})$ in Fig. 1(b) is obtained.
- (3) Rotary table 1 is rotated by β , and rotary table 2 is rotated until the angle sensor reading approaches the initial value M_L and M_{Li+1} is obtained.
- (4) Steps 2 and 3 are repeated N times to yield the following equations:

$$B_{\alpha+i\beta} + \alpha - A_{i\beta} - B_{i\beta} = M_i / S_0 \quad (1)$$

where $M_i = M_{Hi} - M_{Li}$ ($i = 0$ to $N-1$), S_0 is the mean sensitivity of the angle sensor, $A_{i\beta}$ and $B_{i\beta}$ are the dividing errors of i th part of N equal divisions of 2π by rotary tables 1 and 2, and $B_{\alpha+i\beta}$ is the dividing error while the rotary table 2 is rotated by α from $i\beta$.

Since the total sum of $A_{i\beta}$ and that of $B_{i\beta}$ cancel out, the total sum of each term in the equation (1) can be reduced to the following equation:

$$S_0 = \frac{1}{\alpha + (\sum B_{\alpha+i\beta})/N} \times \frac{1}{N} \sum_{i=0}^{N-1} M_i \quad (2)$$

Thus, the mean sensitivity of an angle sensor can be obtained from the mean value of N readings by the angle sensor. Although the calibration error is brought by the term $(\sum B_{\alpha+i\beta})/N$ in equation (2), this error can be decreased with the mean value of N outputs of the equally sampling in a circle of the rotary table. This influence of the dividing errors can be omitted if N is large enough, that is to say, $(\sum B_{\alpha+i\beta})/N$ becomes much smaller than α .

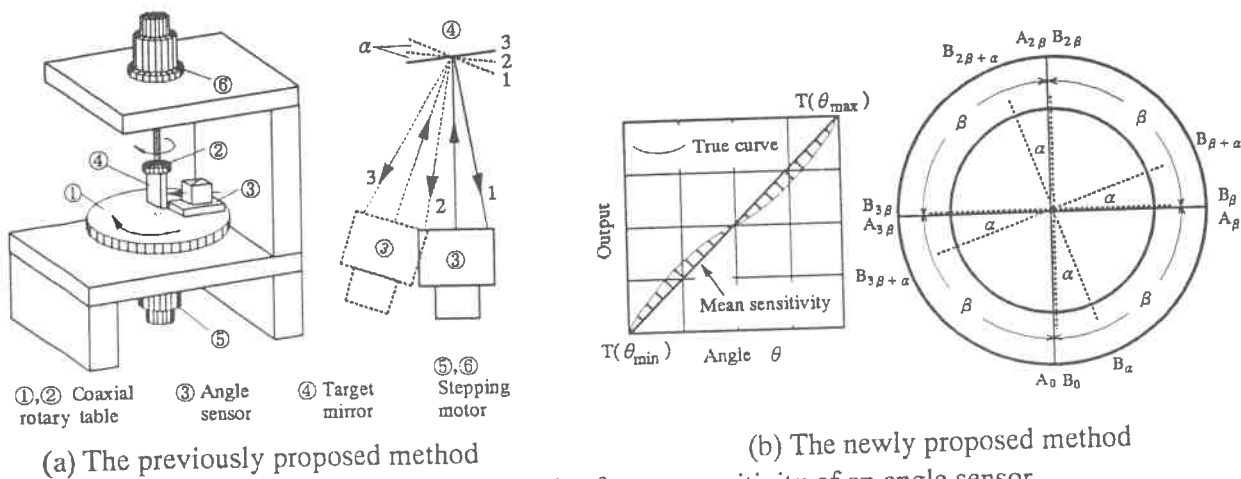


Fig. 1 Calibration methods of mean sensitivity of an angle sensor

3. EXPERIMENTAL RESULTS

The measurement range α of the angle sensor was $500 \mu\text{rad}$ [1]. The measurement interval β was 12.96 mrad which corresponds to a division value based on $N=500$.

Figure 2 shows the thermal drift of the angle sensor. The angle sensor is placed as in the case of the calibration of the mean sensitivity, and the two motors were switched on and maintained in a stationary state as instructed by the computer. The measurement time is approximately 2 hours, and the sampling

period is approximately 4 seconds. The temperature variation during measurement time is approximately 0.2°C . The diagram of the low frequency drift was obtained by taking the moving averages of seven points within the drift curve, while the diagram of the high frequency drift was obtained by subtracting this low frequency drift from the original drift curve. During a period of 2 hours, a low frequency drift (over 5 seconds in period) of approximately 400 nrad is present. In the actual calibration of the mean sensitivity of the angle sensor, the output represents the difference between the maximum value and minimum value in the measurement range is used. And the time to obtain the difference is short (approximately 5 seconds), so that the calibration accuracy is not influenced by low frequency drift. The high frequency drift in Fig. 2 is approximately 30 nrad. The stability and resolution of the angle sensor in the mean sensitivity calibration environment are estimated from the high frequency drift.

Figure 3 shows the variation in the difference obtained by subtracting the average from each reading M_i and the temperature variation in the calibration environment. This system requires approximately 2 hours for the recording of data from a single calibration, and the temperature variation during this time is approximately 0.2°C . The variation of M_i is approximately $30\text{ }\mu\text{rad}$. The main cause of this variation is thought to be lack of divisional accuracy of the stepping motors. Nevertheless, this variation does not remarkably affect the calibration results of mean sensitivity since it decreases with the total sum of the 500 dividing errors per rotation of the stepping motors. The calibration results of the mean sensitivity obtained for four separate rotations and the deviation as determined by the difference between the calibration result and the average value of four results, are shown in Table 1. The stability of the mean sensitivity can be determined using standard deviation $\sigma_s = 0.22 \times 10^{-6} / \mu\text{rad}$, that is, $\pm 50\text{ nrad}$ maximum error is caused by this calibration error when the measurement range is $500\text{ }\mu\text{rad}$.

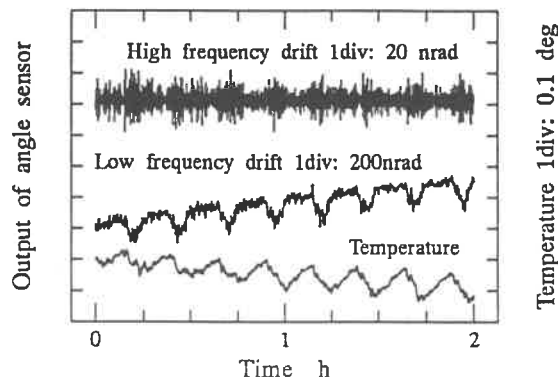


Fig.2 Drift of calibration of mean sensitivity

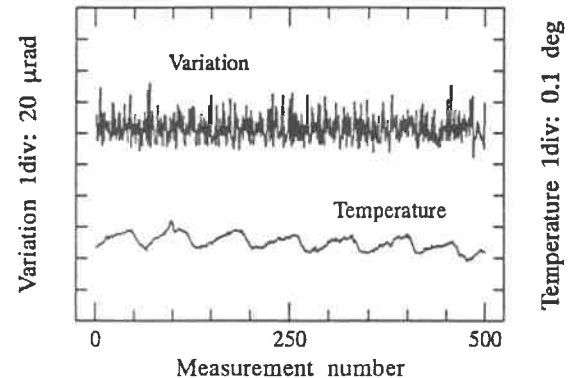


Fig.3 Variation of mean sensitivity

Table 2 shows the average values of S_0 in Table 1, the standard deviation of S_0 and the measurement time when the measurement interval β is 32.4 mrad ($N=200$), 16.2 mrad ($N=400$) and 12.96 mrad ($N=500$), respectively. Although the measurement time increases as the measurement interval β decreases, the standard deviation of the calibration result also decreases. In addition, the influence of the dividing error $B_{\alpha+i\beta}$ can be decreased by averaging the N outputs of the equally sampling in the circumference of the rotary table.

If the angle sensor with $500\text{ }\mu\text{rad}$ measurement range is calibrated by the previously proposed method, $N \approx 10^4$, it is almost impossible to calibrate the angle sensor because the calibration time is

too long. The newly proposed method is not limited by the measurement range of an angle sensor.

For calibrating the linearity error, the average value of calibration results of the four times in Table 1, $1.12554 \times 10^{-3} / \mu\text{rad}$, was used as the mean sensitivity of the angle sensor.

Figure 4 shows the stability of the calibration results of the linearity error of the angle sensor by use of in situ self-calibration method [1]. This figure shows the average value of four calibration results and the difference between the two average values calibrated on different days. The maximum deviation in the difference was approximately 35 nrad. The low-frequency component is approximately 20 nrad and the high-frequency component is approximately 15 nrad in p-p value. The root mean square error of the difference was approximately 10 nrad.

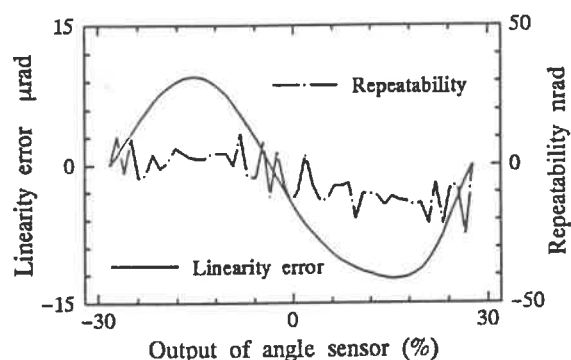


Fig.4 Repeatability of calibration of linearity error

Table 1 Mean sensitivity of the developed angle sensor

Number	$S_0: 10^{-3} / \mu\text{rad}$	Deviation of $S_0: 10^{-6} / \mu\text{rad}$
1	1.12579	0.25
2	1.12533	-0.21
3	1.12566	0.12
4	1.12538	-0.16
Average	1.12554	$\sigma_s = 0.22$

Table 2 Mean sensitivity of the developed angle sensor

$\beta: \text{mrad}$	N	Time: min	$S_0: 10^{-3} / \mu\text{rad}$	$\sigma_s: 10^{-6} / \mu\text{rad}$
32.40	200	50	1.12573	0.34
16.20	400	100	1.12544	0.26
12.96	500	120	1.12554	0.22

4. CONCLUSIONS

The results of the present research can be summarized as follows:

- 1) A new method for calibrating the mean sensitivity of a precision angle sensor with a small measurement range was proposed. The effectiveness of the method was also confirmed.
- 2) An angle sensor which has a measurement range of $500 \mu\text{rad}$ was absolutely calibrated by the new method and the in situ self-calibration of the linearity error of an angle sensor.
- 3) The maximum repeatability error for calibrating the mean sensitivity was approximately $\pm 0.01\%$ of the measurement range, and that for calibrating the linearity error was approximately 35 nrad.

ACKNOWLEDGMENTS

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A METHOD FOR THE MEASUREMENT OF NONLINEARITY IN HETERODYNE INTERFEROMETRY

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Introduction

Optical heterodyne interferometry forms the basis of metrology and control in numerous high precision applications. In polarization encoded laser heterodyne interferometry, the input beam consists of two mutually orthogonal linearly polarized beams of slightly different frequencies. In an ideal interferometer, the two beams separate completely and travel in pure form in the two arms of the interferometer. On recombination, the optical path difference between the two paths is carried by the heterodyne signal as phase relative to some reference. This heterodyne signal has a frequency equal to the difference in frequencies of the input beams. In practice, these beams are contaminated by the leakage of a small quantity of one beam into the path of the other. This phenomenon is known as beam mixing and can be attributed to imperfect components, elliptically polarized input beams and misalignment. While the attainable resolution has steadily become higher with improved signal processing techniques, the achievable accuracy is limited by nonlinearity due to beam mixing to about 5 nm.

The nonlinearity was first predicted by Quenelle [1] and demonstrated experimentally by Sutton [2] who measured 5 nm peak-to-peak nonlinearity using the technique of pressure scanning. In addition to the nonlinearity with a periodicity of one cycle per 2π change in optical path length predicted by Quenelle, he also observed a smaller nonlinearity with a periodicity of two cycles over the same optical path length change (hereafter referred to as first-harmonic and second-harmonic nonlinearity respectively). Subsequent theoretical analyses have examined various aspects of this problem [3-9]. Experimental investigations using different techniques [10-18] have also been performed. These include the use of a reference interferometer in combination with mechanical magnification [14,15], piezo-electric translators with integral capacitance metrology [13], measurement of the amplitude modulation of the beat signal [12] and the technique of pressure scanning [18]. Each of the foregoing methods suffers from some drawbacks. The pressure scanning method requires the use of a pressure chamber which is extremely inconvenient for large systems. In addition, this requires the use of a pressure sensor of sufficient linearity and a mechanism to vary the pressure in a controlled fashion. Pressure scanning is also a slow process and is as such unsuitable for *in situ* diagnostics of interferometer systems. Techniques using a second reference interferometer require complex metrology and magnification by suitable methods to overcome the nonlinearities of the reference interferometer. Techniques using piezo-electric translators suffer from the same limitations as systems with reference interferometers. The amplitude modulation technique is free from the drawbacks of the above methods and can be used as a *in situ* diagnostic. However, this method is unable to discriminate between the first and second harmonic nonlinearities and is limited by the dynamic range of the instrument in use for the simultaneous measurement of small nonlinearity signals superimposed on a large base signal.

This paper shows how the technique of velocity scanning may be used to perform *in situ* diagnostics of a interferometer system and aid in the optimization of alignment. This method was first described by Patterson and Beckwith [17]. The theoretical basis for this method is described and is followed by experimental results that show the application of this technique in the determination of the influence of various parameters such as PBS rotational alignment, analyzer orientation and retroreflector azimuthal position on nonlinearity in a two-retroreflector displacement measuring interferometer.

Theory

The source of beam mixing nonlinearity can be seen in the simple interferometer depicted in Figure 1. The input beam from a two-frequency laser source consists of two mutually orthogonal nominally linearly polarized beams of slightly different frequencies, ω_1 and ω_2 . In the ideal situation, the incident beams are free from ellipticity and are completely separated by the polarizing beamsplitter (PBS). However, due to various factors, incomplete separation of the beams results in mixing, with the consequence that some amount of the beam intended for the measurement arm finds its way into the reference arm and vice-versa.

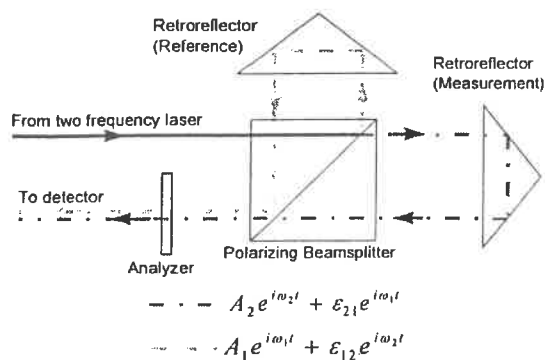


Figure 1 Simplified heterodyne interferometer with mixing in both arms

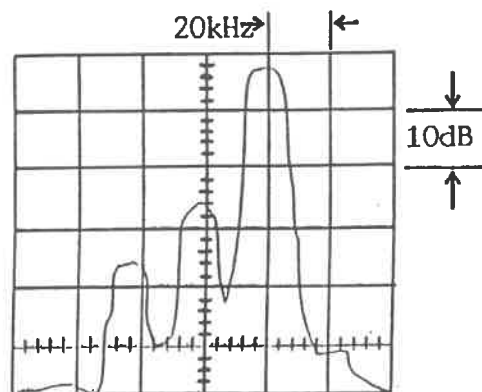


Figure 2 Typical spectrum analyzer output showing the three frequencies for $v=7$ mm/sec approx.

These leakage quantities are represented by ϵ_{12} and ϵ_{21} respectively. Neglecting DC and quasi-DC terms and after normalizing by the nominal signal, the resulting intensity upon recombination, I_{rel} , as seen at the photodetector may be expressed by

$$I_{rel} \approx 2\Re\{(\Gamma_0 e^{i(\Delta\omega t + \Delta\phi)} + \Gamma_1 e^{i\Delta\omega t} + \Gamma_2 e^{i(\Delta\omega t - \Delta\phi)})\} \quad (1)$$

where $\Delta\omega = \omega_1 - \omega_2$ and $\Delta\phi$ represents the change in optical path length due to motion of the target retroreflector and is related to the physical path length change Δs by:

$$\Delta\phi = \left(\frac{2\pi}{\lambda}\right) \Delta s = k\Delta s \quad (2)$$

where λ which is the wavelength of the light in the medium that the interferometer is operating in and replaces λ_1 and λ_2 which are taken to be nearly equal. Γ_0 represents the base signal and in this simplified model is equal to unity. Γ_1 and Γ_2 represent the periodic error terms. Γ_1 and Γ_2 are given by

$$\Gamma_1 = \frac{\epsilon_{21}}{A_1} + \frac{\epsilon_{12}}{A_2}, \quad \Gamma_2 = \frac{\epsilon_{12}\epsilon_{21}}{A_1 A_2} \quad (3)$$

In the absence of beam mixing Γ_1 and Γ_2 vanish. If Γ_1 and Γ_2 are small, they result in phase errors with amplitudes equal to their magnitude and with periodicities of one and two cycles per 2π of optical path length change respectively.

If the measurement retroreflector is moved at a constant velocity v such that $\Delta s = Nvt$, then substituting in Eq. 2, $\Delta\phi$ becomes

$$\Delta\phi = Nkv t \quad (4)$$

and N is an integer that depends on the number of passes in the interferometer with $N=2$ for this configuration. Substituting in Eq. 1 results in:

$$I_{rel} \approx 2\Re\{(\Gamma_0 e^{i(\Delta\omega + 2kv)t} + \Gamma_1 e^{i\Delta\omega t} + \Gamma_2 e^{i(\Delta\omega - 2kv)t})\} \quad (5)$$

The three terms in Eq. 5 produce three distinct sinusoidal signals of relative magnitudes Γ_0 , Γ_1 and Γ_2 and frequencies of $\Delta\omega + kv$, $\Delta\omega$ and $\Delta\omega - kv$ respectively. These signals may be observed directly at the output of the photodetector. Figure 2 shows one such typical output.

Experimental Setup

The setup consists of a two-frequency laser head with a 20 MHz frequency split. The laser beam enters the PBS which has the reference retroreflector bolted directly to it. The target (measurement) retroreflector is mounted on a motorized stage which can be moved at a constant velocity. The retroreflector itself is mounted in a rotary stage that allows for the adjustment of the azimuthal position of the retroreflector about an axis parallel to the input and output beams and midway between them. The PBS/reference retroreflector assembly is mounted in a similar rotary stage that allows for azimuthal adjustment about nominally the same axis as the target retroreflector. The retroreflectors are made of glass and are silvered on the reflecting surfaces. The output beam passes through a Glan-Thompson polarizer (hereafter referred to as the analyzer) into a fiber-optic coupler that launches the beam into an optical fiber. The optical signal is converted into an electrical signal by a remote photodetector. The electrical signal is fed to a Tektronix 7L12 RF

Spectrum Analyzer. The stage is driven at a velocity of approximately 11.3 mm/s which results in a frequency separation of approximately 35.7 kHz.

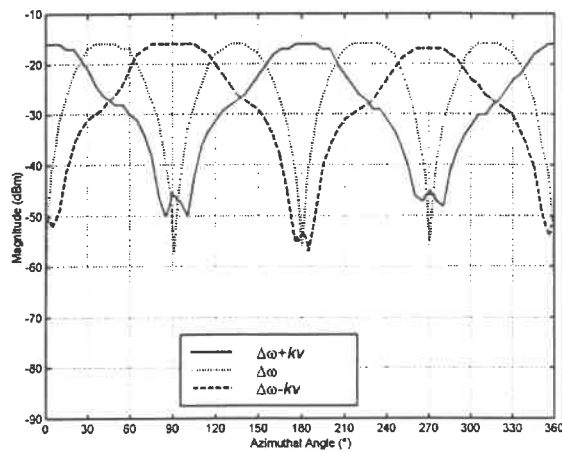


Figure 3 Variation of the magnitude of the three components with PBS azimuthal orientation

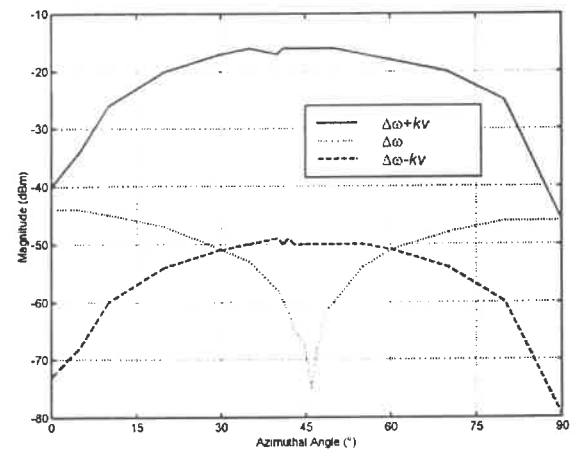


Figure 4 Variation of the magnitude of the three components with analyzer azimuthal orientation

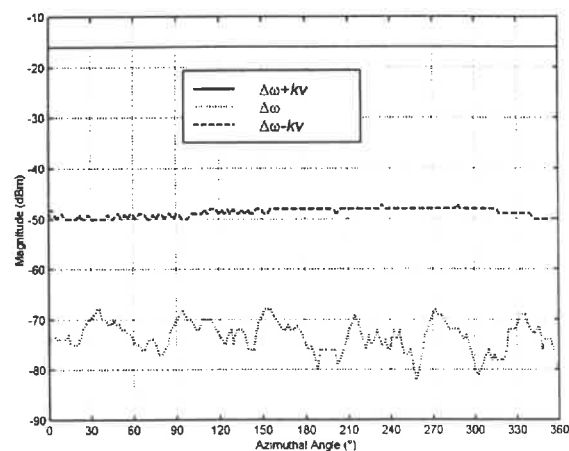


Figure 5 Variation of the magnitudes of the three components with measurement retroreflector azimuthal orientation

of the edges of the retroreflector appears horizontal when looking into it) was established as the zero orientation. All the magnitudes are recorded at the same point in the travel of the stage while the stage is moving away from the interferometer.

The effect of PBS rotational alignment was determined by rotating the PBS through 360° and recording both the azimuthal angle and the magnitudes of the three frequency components. The results are shown in Figure 3. It can be seen that first-harmonic nonlinearity approaches a minimum at approximately 0, 90 and 270 degrees. The base signal and the second-harmonic nonlinearity (which occur at $\Delta\omega+kv$ or $\Delta\omega-kv$ depending on which frequency is dominant in the measurement arm) are also seen to vary in magnitude.

The effect of analyzer rotation was determined using a similar procedure. The PBS was positioned at 0°. The results are shown in Figure 4.

The effect of retroreflector orientation was determined by rotating the retroreflector in 2° increments over 360°. The PBS and analyzer are positioned at 0° and 45° nominally. Figure 5 shows that retroreflector rotation has negligible

Experimental Technique and Results

One of the applications of this technique is the generation of data for the verification of analytical models of the system. To this end, the zero reference angle for both the analyzer and polarizer azimuthal angle need to be established. In the case of the PBS, this was set by directing the beam in the measurement arm directly onto a photodetector. The azimuthal position of the PBS was then adjusted and the position of the PBS that minimized the magnitude of the 20 MHz beat signal was established as the zero position. In the case of the analyzer, the PBS was first adjusted to the zero position and the beam in the reference arm was then passed through the analyzer onto a photodetector. The zero position of the analyzer was established by minimizing the 20 MHz signal as before. This procedure eliminates the effect of other optics in the system on the azimuthal orientation of the plane of polarization of the input beams and consequently on the zero reference position. In the case of the measurement retroreflector, an arbitrary position (oriented such that one

effect on the base signal and the second-harmonic nonlinearity. The variation in the first-harmonic nonlinearity shows six-fold symmetry and illustrates the sensitivity of this technique to small variations in nonlinearity.

Conclusions

The application of the velocity scanning technique to the rapid assessment of interferometer nonlinearity has been described. This method does not require an external reference such as a pressure gage in the pressure scan technique or another independent displacement transducer against which nonlinearity may be compared. The various frequency components may also be determined directly. It has also been shown that an interferometer may be 'tuned' using this method to obtain maximum performance. The sensitivity of first-harmonic errors to analyzer alignment has been demonstrated and is in agreement with results previously reported by the authors [18]. It is also clear that the effects of retroreflector azimuthal orientation are negligible when compared to the effects of angular misalignment of either the PBS or the analyzer. The applicability of the technique extends to all systems except those where the target cannot be moved at a velocity sufficient for adequate frequency discrimination of the three signals.

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Functional Digital Filtering for the 3-D Topography Measurement Using a Confocal Scanning Laser Microscope

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Key words: scanning laser microscope, digital filter, roughness parameter, stylus instrument

1. Introduction

Unreasonable short-wavelength components tend to be included in the raw data of surface texture measured by a confocal scanning laser microscope (SLM). These short-wavelength components can be attributed to the intrinsic overshoot of SLM at steep slope on an object surface [1]. In order to obtain proper surface topography data, it is necessary to attenuate these short-wavelength components functionally.

A research has been made to improve the validity of the topography data by SLM [2]. From a practical point of view, authors introduced a functional digital filtering process to eliminate such short-wavelength components. A design procedure in spatial frequency domain was developed, which is based on the spectrum amplitude ratio of the

topography data acquired by both SLM and a stylus instrument (STYLUS).

2. Filter design

Figure 1 shows the representative 3D contour plots of measured topography data for an Electro Discharge Machined (EDM) surface. It is clear that more short-wavelength components are included in the (a) SLM data than in the (b) STYLUS data. In this study, these short wavelength components are assumed to be random and removed by using a certain digital filter. A basic process of the filtering is shown in Figure 2. In the first place, topography data of the specimen surface are measured by both SLM and STYLUS. Then, transmission characteristic curve of the filter is derived from the respective

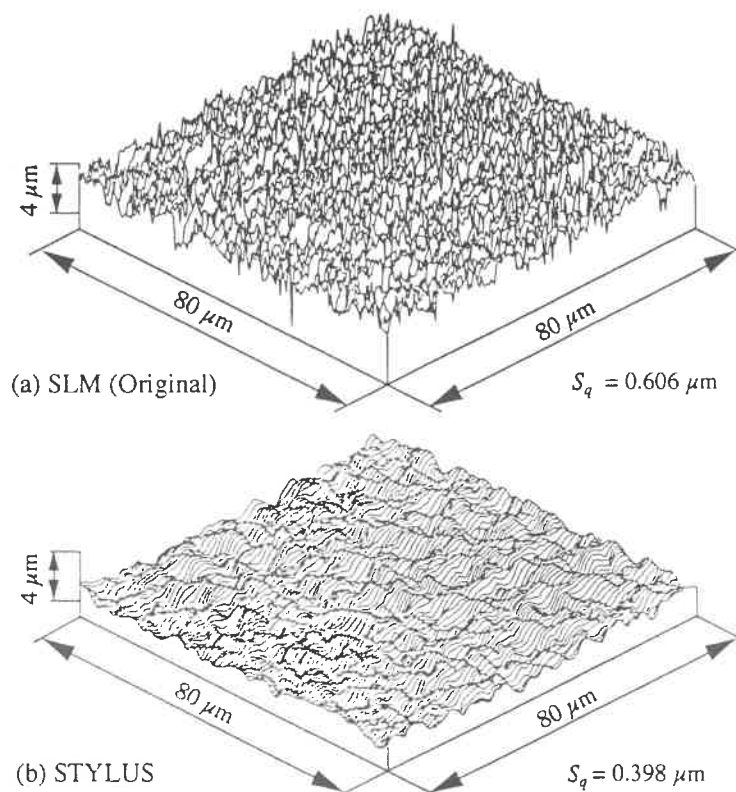


Fig.1 3-D plots of measured topography data (EDM)

ratios of the spectrum amplitude of those two 3D topography data. The filter characteristics $H(\omega_x, \omega_y)$ can be obtained by a so-called inverse filtering method. Finally, the filter is applied to the original SLM data to obtain more reasonable topography data.

Figure 3 shows the radial power spectra of the EDM surface $Z_{iM}(\omega_r)$ and $Z_{oM}(\omega_r)$, by SLM and STYLUS respectively. These 1D spectra were obtained by averaging 2D spectrum in the circumferential direction. It is noted that $\omega_r^2 = \omega_x^2 + \omega_y^2$. The plot shows that the SLM data contains much more high frequency component than the STYLUS data. We introduce the ratio of Z_{oM} to Z_{iM} and express by R_M .

$$R_M = \frac{Z_{oM}}{Z_{iM}} \quad (1)$$

However R_M undulates in the low frequency region as shown in Figure 4. It is preferable that the filter characteristic curve is rather smooth. Then H_M was represented as a following approximation function of ω_r with two variables.

$$H_M(\omega_r) = \frac{1}{\sqrt{1 + \left(\frac{a}{\omega_r^b}\right)^2}} \quad (2)$$

Coefficient a relates to the signal to noise ratio. Coefficient b

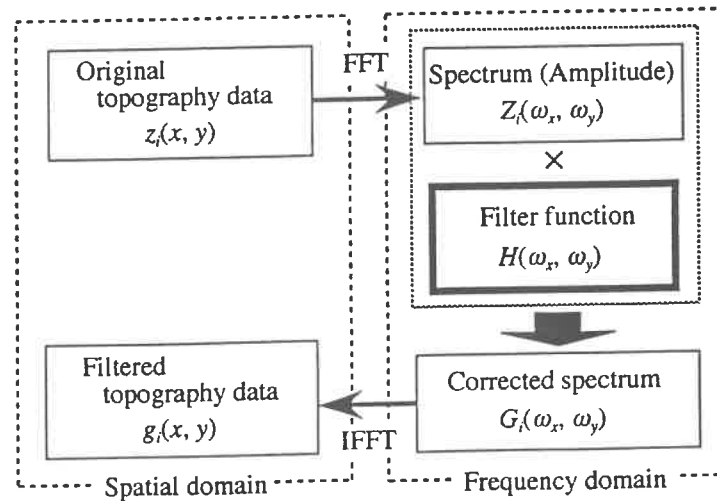


Fig. 2 Filtering process

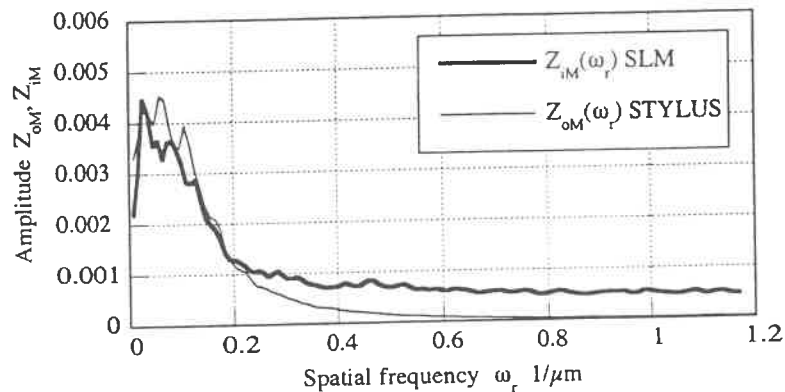


Fig. 3 Comparison of the averaged spectra between SLM and STYLUS

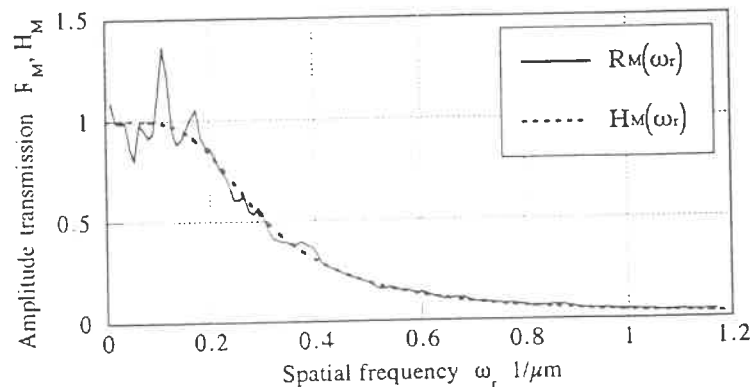


Fig. 4 Design procedure of the functional filter

relates to the coefficient of the power law of the spectrum. This approximation function was fitted to R_M by the iterating least square method. This 1D spectrum $H_M(\omega_r)$ was substituted for the 2D filter $H_M(\omega_x, \omega_y)$ by applying it in all the direction.

3. Result of filtering

A contour plot of the corrected data using this filter is shown in Figure 5. It is apparent that the filtered SLM data looks very similar to the STYLUS data shown in Figure 1(b).

Six parameters, S_q , S_{sk} , S_{sq} , S_{al} , S_{ds} and S_{sc} , from a set of primary 3-D topography parameter [3] were introduced to discuss the similarities between the filtered SLM data and the STYLUS. An empirical data processing technique was employed to avoid the influence of sampling interval and high frequency component when calculating S_{ds} , S_{sq} and S_{sc} [4]. EDM, ground and two types of wet-blasted (alumina abrasive, glass beads abrasive) surfaces were put under test. At least five sample surfaces were provided for the each machining method. One filter was designed individually for each machining method by using the surface that has the median in amplitude parameter. Differences in the roughness parameters S_q , S_{al} and S_{sq} between the filtered SLM data and the STYLUS were smaller than 15%. The effectual range of the filter was proved to be wide enough for practical use. However parameters such as S_{sk} , S_{ds} and S_{sc} have considerably large differences. It is because that these parameters are sensitive to the phase shift between each frequency component, which is not considered in this filtering method.

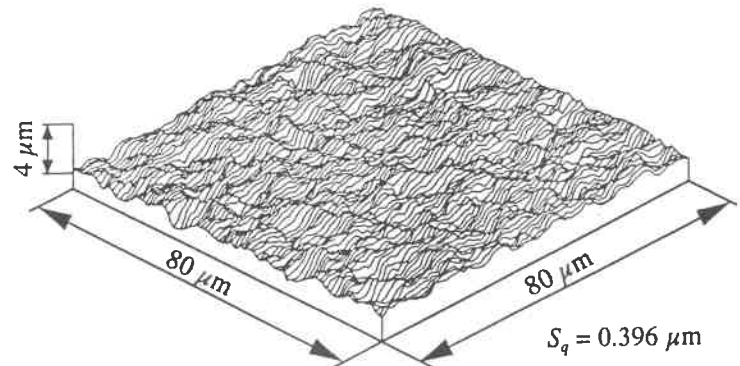


Fig. 5 Functionally filtered topography data (EDM)

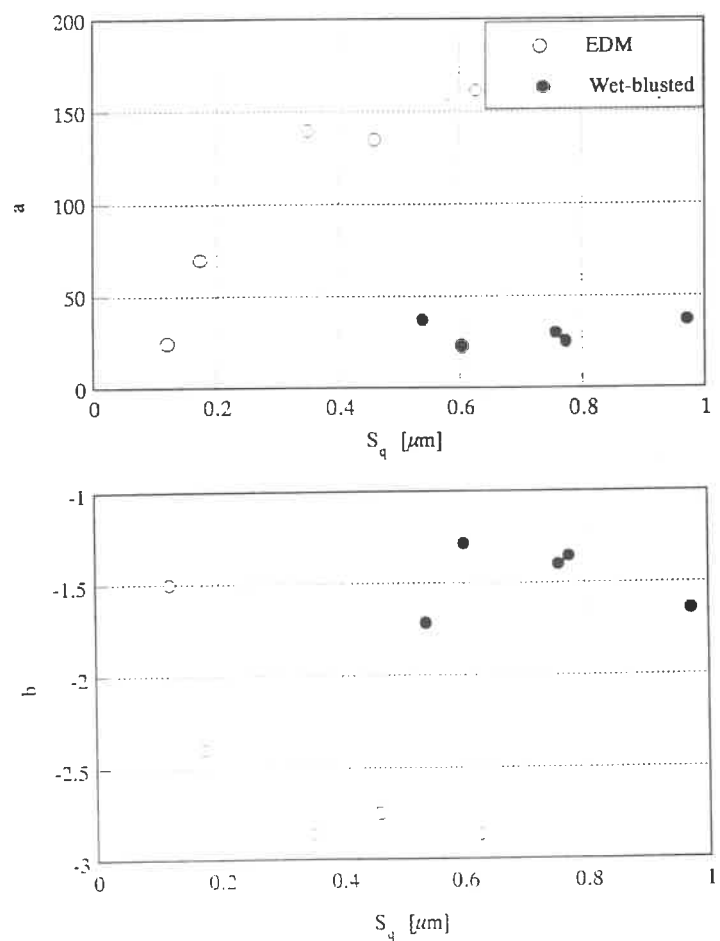


Fig. 6 Relation between S_q and Filter characteristics parameter a, b

4. Dependence of the functional filter on surface texture

The dependence of the filter function on the surface texture and its machining method were examined. Figure 6 shows the variation of the filter variables a , b with S_q value and the machining method. The wet blasted surfaces in this study were finished by the same abrasives, but by changing only the jet velocity at

nozzle outlet. It is apparent that the single filter function can be used for these wet blasted surfaces. It can be concluded that it is because lateral surface wavelength changes little while vertical amplitude changes more. In contrast a , b for EDM surfaces change significantly. In the case of the EDM surfaces, machining factor was the discharge voltage, which may lead to the change in both lateral wavelength and vertical amplitude of the surface. It should be noted that a further study is necessary to reveal appropriate filter characteristics for the EDM surfaces.

5. Variation of the filter due to objective N.A. and

Figure 7 shows variation of the amplitude spectra with numerical aperture (N.A.) of the objective. There are more high frequency components included in the data by smaller N.A. than that of the larger N.A.. This phenomenon may be attributed to the fact that the objective of small N.A. value may not detect specular reflection from steep facets. Therefore the filter should be designed in consideration of the characteristics of the optical system of the employed SLM.

6. Conclusion

- (1) A Design procedure of the functional filter to obtain valid 3D topography data from the data measured by the confocal scanning laser microscope is developed.
- (2) Effectiveness of the filter is indicated using 3D roughness parameters.
- (3) The filter characteristics are dependent on surface texture, machining method and characteristics of the optical system of SLM.

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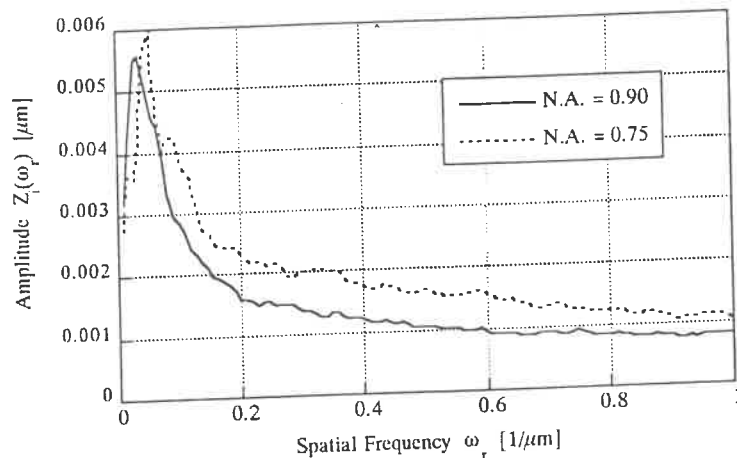


Fig.7 Variation of the spectra with different objective N.A.

Hysteresis compensation of AFM by Preisach model

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1. Introduction

In equi-force mode among the AFM scanning modes, the scanning data of a sample is acquired by Z PZT input voltages⁽¹⁾. But, PZT actuator has a hysteresis nonlinearity between the input voltages and the output displacements. The hysteresis is a major limitation of AFM's resolution. Closed loop control of PZT actuators can eliminate the effect of hysteresis. However, in AFM system with laser deflection method⁽²⁾, it cannot be available by two reasons. First, it's not easy to instrument the Z displacement sensor. Second, the most important reason, the Z displacement sensor with below 0.01nm resolution does not exist. So we need to compensate scanning data without displacement sensor. Several compensation methods such as capacitance insertion method⁽³⁾, feed-forward control method⁽⁴⁾ have been suggested. But these methods could not eliminate the hysteresis completely. Lately, preisach model developed to represent the hysteresis behavior of ferromagnetic materials mathematically has been applied to PZT hysteresis modelling and proved to be a very good modelling tool for PZT hysteresis.⁽⁵⁾⁽⁶⁾ Because preisach model reflects a input history in deducing the displacements from the input voltages, it can represent the PZT hysteresis more accurately than any other methods. We adopt classical preisach model to compensate for the PZT actuators in AFM stage. This can be done by extracting the surface profile data from the PZT input data which has been compensated by Preisach model. This method doesn't need any displacement sensors and is very accurate than any other methods. We compensate scanning data by classical preisach model and compare it with a raw data.

2. Classical Preisach Model for hysteresis

The mathematical form of the classical preisach model can be written as follows :

$$x(t) = \int \int_{\alpha \geq \beta} \mu(\alpha, \beta) \gamma_{\alpha\beta}[u(t)] d\alpha d\beta \quad (1)$$

where $x(t)$ is the output displacement of a PZT actuator and $\mu(\alpha, \beta)$ is a weighing function that can be acquired by the experiments. α, β correspond to "up" and "down" switching values of the input voltages. $\gamma_{\alpha\beta}[u(t)]$ is a hysteresis operator whose value is determined by the input history. It's proven experimentally that the PZT actuator(AE0505D08, TOKIN) used in our AFM system has a wiping-out property and congruency property-necessary and sufficient conditions for a hysteresis actuator to be represented by the classical preisach model.⁽⁵⁾ And we use the explicit form in compensating the scanning data.⁽⁶⁾

3. Experimental Setup

Scanning experiments were performed using the AFM system that has been made in our laboratory shown in fig. 1. The head and XYZ scanner are main components of the AFM. The head has a flexible hinge mechanism to align the laser ,tip and PSD which would increase the head stiffness and noise removal rate. The XYZ scanner has monolithic and XYZ-decoupled structure with flexible hinge mechanism which would, also, increase the noise removal rate.

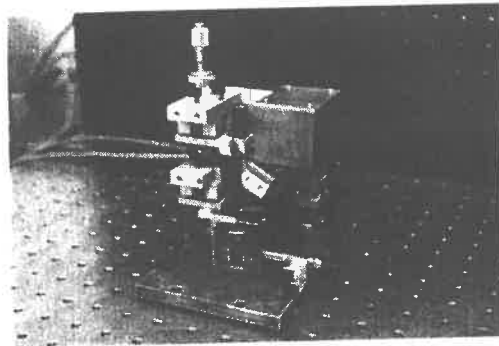


Fig. 1 AFM system

4. Compensation of scanning data by the classical preisach model

Scanning experiments were performed with the silicon standard sample shown in Fig. 2. It was scanned by a method that XY moving step was 2 Volt(nominal length of PZT is 120nm) and scanning frequency was 3 Hz, that is, 0.33 sec was needed to get a height data at one point. Generally, PZT has creep property-changing it's length in time with a constant input voltage. To avoid the creep effect, fixed scanning time(0.33 sec) corresponding to the acquisition time for weighing function for classical preisach model was applied.

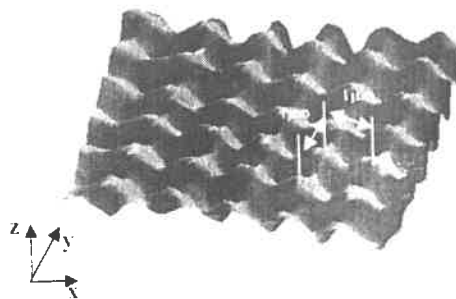
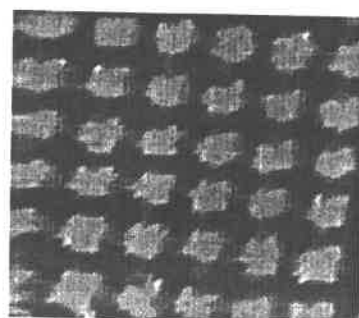
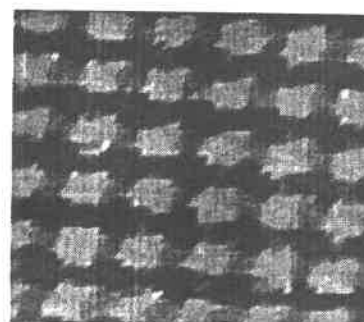


Fig. 2 Standard test Sample

Fig.3 shows two scanning images; one is the raw data, that is, input voltages are mapped linearly to PZT displacements(100V to $6\mu\text{m}$) and the other is compensated data by the classical preisach model. Compensation was made in three axis. Especially, Fig.3 shows a compensation effect in X, Y axis. In Images, the white area is a peak and the black area is a valley. It is shown that, in compensated data, distortion decreases dominantly in vicinity of the edge. The height data in one line along the x-axis is displayed in Fig. 4. It shows that in the compensated data, peak-to-peak length is shorter than the raw data. It can be explained as follows : If AFM tip moves from the valley to the peak of sample and moves again from that peak to the other side valley like shown in Fig. 5(a), the monitored voltages entering to the Z PZT which moved the tip is shown in Fig. 5(b). If we assume the input voltages are mapped to output displacements linearly, the trajectory is like b-c-b. But if we take the PZT's hysteresis property into account, the trajectory is like a-c-d which has difference between the point a and the point d. In Going and returning trajectory, real profile of the surface has shorter peak-to-peak length than one produced by the pure monitored voltage(linearly mapped to displacements). Since preisach model can compensate this effect, the compensated data by the preisach model has shorter peak-to-peak length than the linearly mapped data.



(a) Raw scanning data



(b) Compensated Scanning data

Fig. 3 Comparison between the raw data and the compensated data

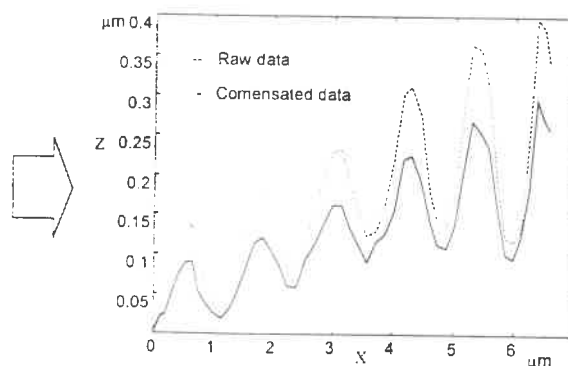
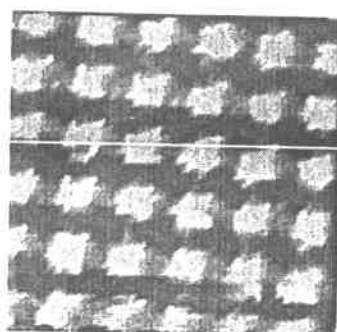


Fig. 4 Z axis scanning data comparison

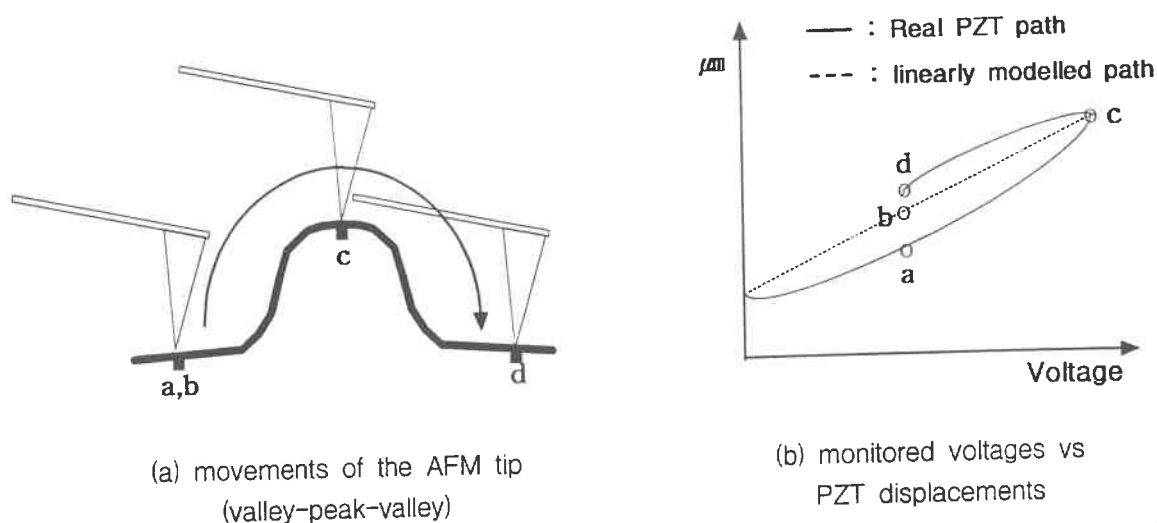


Fig. 5 Hysteresis effect on Z scanning data

5. Conclusion

Classical preisach model is applied to AFM scanning data. It's been shown that AFM image is improved by the preisach model compensation. We examined why the Z profile has such distortion when the monitored input voltages are mapped linearly to output displacements and suggested that such a distortion can be cured by the classical preisach model.

ACKNOWLEDGEMENTS

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Simulation of a Cylindrical Interferometer for Comparison to Stylus Measurement Techniques

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Abstract

A cylindrical interferometer developed by Tropel [1] has the ability to measure cylindrical and tapered parts for cylindricity, roundness, straightness, and other metrological attributes. A simulation was performed to relate the cylindrical interferometer measurements to traditional stylus-based cylindrical and roundness measurement techniques. Using the software package Matlab, a stylus instrument was modeled convolving a filter function simulating a stylus with a pink noise surface. The cylindrical interferometer was also modeled with an effective interferometer filter function. Roundness results for each stylus trace are shown. Also roundness results averaging corresponding data points in successive stylus traces were shown to converge to the roundness results simulated with the cylindrical interferometer.

1. Introduction

Of particular interest concerning a cylindrical interferometer [2,3,4] is whether the roundness of a cylindrical part measured by this type of interferometer is the same as or different from the roundness measured by a stylus instrument. Since the Tropel CM25 is an optical metrology instrument that incorporates grazing incidence interferometry and fringe pattern analysis for data acquisition while a stylus performs a mechanical measurement over a small contact patch, roundness measured with each instrument may be different because of the inherently different methods employed by each instrument.

In this paper, both a stylus and a cylindrical interferometer are simulated using the Matlab software package. A surface representing a band of data points around a cylinder is generated that is sampled by both the simulated stylus and the simulated interferometer. Results are shown detailing how measured roundness results both diverge and converge.

2.1 Method of Stylus Simulation

Initially a surface was generated that simulated a strip of surface data from a cylindrical artifact. Within the software package Matlab, an eight bit two-dimensional array was created of a gaussian white noise surface with defined mean and variance. This surface was then convolved in the spatial domain with an averaging filter. The spectral density of the surface after convolution had the form $1/f$. The white noise surface was now "pink" to better model surfaces measured by both a stylus and cylindrical interferometer.

Mathematically, modeling of a stylus has been performed by various means. [5,6,7] For this study, an exact mathematical representation was not used. Since the simulation was to study convergence or divergence of the mechanical and optical measurement, the model used need only be good enough to show to first order how a stylus follows the surface. A stylus tip was modeled as a spatial domain parabolic filter as it moved over the surface. Since the most important factors in forming a stylus trace are the respective geometries of the surface and stylus, the stylus trace was regarded as a convolution of the stylus and surface geometries. [8] The simulated pink noise surface was then convolved with the two

dimensional parabolic filter in the spatial domain for one trace in the azimuthal direction. The total number of stylus traces simulated was a function of the axial spacing between traces. Each stylus trace was then convolved with a 15 undulations per revolution (upr) cutoff gaussian filter with a Nyquist criteria of 4 data points, also in the spatial domain. The output from the gaussian filter gave the roundness trace of the simulated pink noise surface. P-V roundness for this experiment was defined as the maximum data point minus the minimum data point of the roundness trace after a least squares circle was removed from the trace. [9] RMS roundness was also calculated.

2.2 Method of Cylindrical Interferometer Simulation

The cylindrical interferometer was simulated using the same basic concept as the stylus simulation. The interferometer was modeled as a uniform averaging function over the size of one pixel. This "effective stylus" of the interferometer was convolved in the spatial domain over the simulated pink noise surface. This surface was only 1 pixel wide in the axial direction to aid in reducing computational time. The simulated surface was 750 pixels wide in the azimuthal direction based on acquiring data with a 512x480 CCD camera and the interferogram filling the region of interest on the CCD. The output from the interferometer effective stylus convolution with the simulated surface was also filtered with a 15 upr gaussian filter with a Nyquist criterion of four data points. A least squares circle was removed from the single roundness trace resulting from the convolution. Roundness for this one trace was found using the previously defined method.

3. Results

The simulated pink noise surface was an array 8250 data points in the azimuthal direction and 154 data points in the axial direction. With the distance between each data point equal to two micrometer, this corresponding to a simulated cylindrical sample of 5.25 mm diameter and 0.31 mm height. For a 1 mm spherical diameter stylus tip applying 5 g of force onto a surface, a hertzian contact patch of about 10 micrometers was calculated. [10] The stylus filter function was a two dimensional parabolic filter 5x5 data points in size. This filter function was convolved over the simulated surface every 5 data points in the circumferential direction and every row of data points in the axial direction. This resulted in 1650 data points for every stylus trace and a total of 154 simulated stylus traces. Since the height of the surface was set to correspond to one roundness trace for the interferometer, only one roundness trace was simulated from 750 data points. This corresponded to the pixel sampling of 0.31 mm in z and 0.022 mm in r-theta.

The P-V and RMS roundness found from each stylus trace were plotted and an average of all P-V roundness traces was 8.84 micrometers and the average of all 4*RMS roundness traces was 7.04 micrometers. This plot is shown in Figure 1. The cylindrical interferometer's filter function was convolved with the simulated surface and a P-V roundness of 1.18 micrometers and 4*RMS roundness of 0.96 micrometers was calculated. These roundness measurements differ greatly from the average of all the individual stylus traces. Many stylus traces averaged over the spatial surface of one roundness measurement with the interferometer do not result in similar roundness values.

Data points in the first stylus trace were then averaged with corresponding data points in the second stylus trace and a resultant roundness value was determined. The data points in successive traces were averaged until corresponding data points in all traces had been averaged with P-V and RMS roundness calculated for each averaged data set. The plot of averaging data points for successive roundness traces is shown in Figure 2. This plot shows that as a greater number of corresponding stylus trace data points were averaged together, the calculated P-V roundness asymptotically approached a roundness of about 2 micrometers with respect to the square root of the number of stylus traces averaged. The 4*RMS roundness value asymptotically approached about 1.3 micrometers. The P-V roundness somewhat approached the simulated interferometer value of 1.18 micrometers and the 4*RMS roundness definitely approached the interferometer value of 0.96 micrometers.

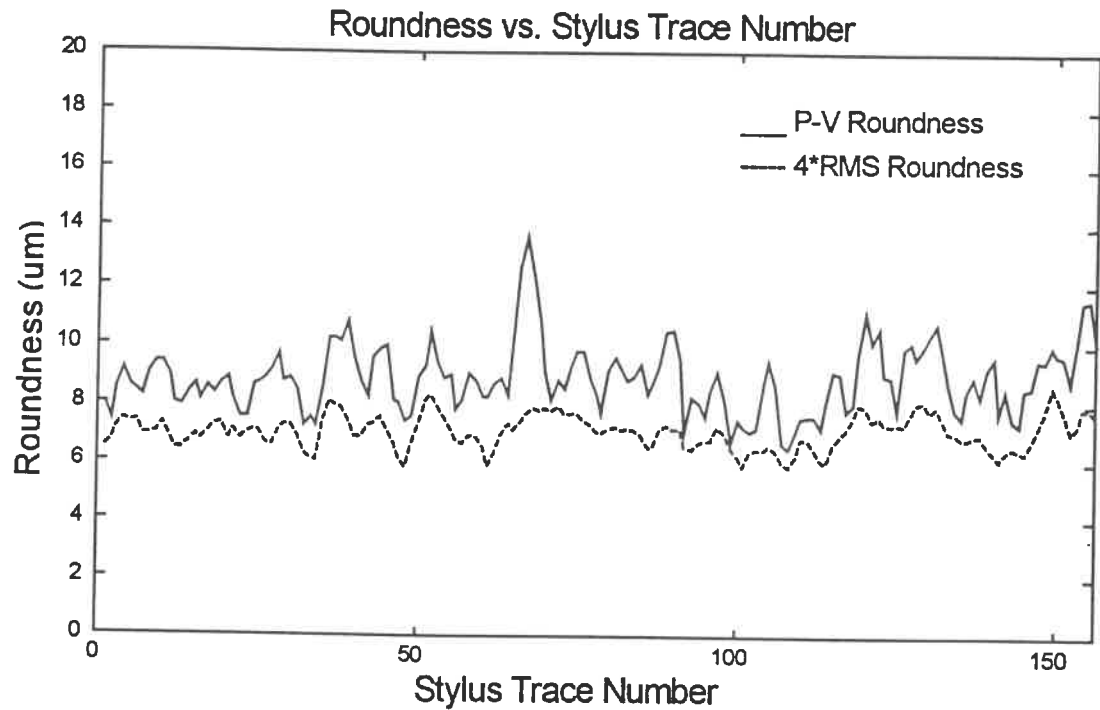


Figure 1: P-V and RMS Roundness of simulated cylindrical surface for 156 stylus traces each displaced 2 micrometers above the previous along the z axis height.

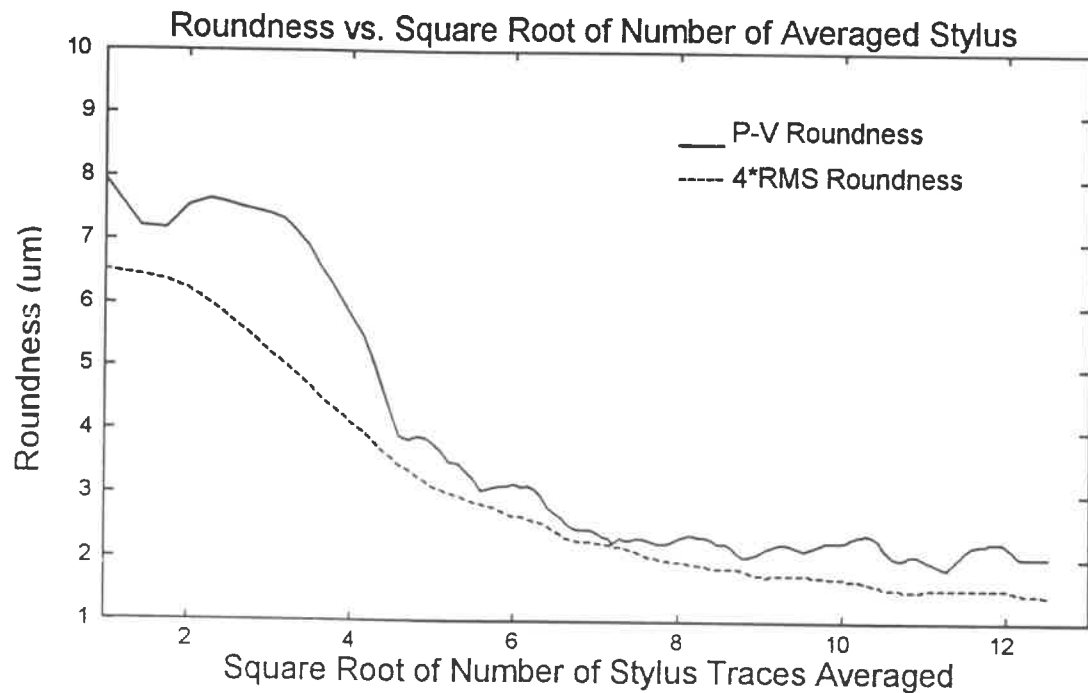


Figure 2: Roundness of surface averaging corresponding data points for successive stylus traces
Simulated cylindrical interferometer P-V roundness was 1.18 micrometers and 4*RMS roundness was 0.96 micrometers.

4. Conclusion

A band from a cylindrical surface was simulated as a two-dimensional set of azimuthal and axial data points. This surface was convolved with filter functions representing a stylus instrument and a cylindrical interferometer with the roundness for each trace calculated. The average of the P-V and RMS roundness found for 154 different, sequential displaced stylus traces was found to differ greatly from the P-V and RMS roundness determined by the one trace of the interferometer over the same data set. When corresponding data points for successive stylus traces each displaced by a constant height from the previous were averaged, the RMS roundness calculated from the resultant averaged trace was shown to converge to a value close to the RMS roundness found by the interferometer as the number of stylus traces averaged increased. P-V roundness from the simulated stylus did not as closely converge to the interferometer P-V roundness when compared to RMS roundness.

Since an interferometer and a stylus instrument sample a roundness artifact with inherently different techniques, it would be expected they give differing roundness measurements. With sufficient averaging of multiple stylus measurements, roundness values can be made to converge to those taken by a cylindrical interferometer.

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FINITE ELEMENT MODELLING OF A ROUNDNESS MEASURING INSTRUMENT

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Abstract The finite element method (FEM) is an effective, efficient and economical way of improving the dynamic behaviour of metrology equipment. This paper presents the first stage of work using the finite element method to improve the dynamic behaviour of a Taylor Hobson Roundness Measuring Instrument. An off-line FE model of the instrument has been developed and the natural frequencies and the corresponding mode shapes have been extracted. An experimental modal analysis was then used to obtain the actual natural frequencies and mode shapes of the instrument. The comparison of the FE results with those from the experimental work has been made, which has demonstrated the validity of the FE model of the instrument.

Key words: Finite element analysis, dynamic characteristics, Metrology instrument

Introduction

Up to the present time, the improvement of the dynamic performance of metrology equipment has relied to great extent on the designers' experience and intuition. However, the finite element method has been applied in a variety of areas. It is desirable to develop a valid FE model for metrology instruments, so that improvement to their design can be undertaken using FE models rather than physical experiments. This process should be more efficient and more economical. The Roundness Measuring Instrument, as shown in Fig. 1, is a leading-edge instrument which has the capability and flexibility to measure various geometrical errors, i.e. roundness, straightness, for a diverse range of components. The instruments are used throughout the world due to their speed, accuracy and powerful software capability. In the following sections, the setting up of an FE model of the Instrument is described. This FE model of the Instrument structure has been verified by experimental modal analysis. The comparison between the dynamic characteristics of the FE model and the instrument has shown the validity of the former. This has created a base for further FE analysis, simulation and optimisation of the instrument.

Definition of the FE model for the instrument

Finite element modelling is a two stage process. In the first stage, the actual structure is replaced by a model of simplified geometry. This eliminates the minor structural details but retains the essential features to be investigated. It would be too costly and time-consuming to model every small detail of the structure. The second stage is to represent the simplified model by a finite element model.

a. Setting-up the geometry model for the instrument

The geometry model of the instrument consists of nine primary components, including: the arm, carriage, column, column base, upper and lower parts of the column spherical joint, base insert, epoxy concrete base, and spindle. Parts of the instrument that are not included in the geometry model are bolts, electronic parts, wires, the steel frame, four rubber supports, the aluminium covers, belts and other accessories.

b. Mesh generation

The models of those components whose geometry is simple (the arm, the carriage and the column) were meshed using 8-noded brick elements. All of the other components were meshed by using 4-noded tetrahedral elements, that is, C3D8 and C3D4 in ABAQUS. The mesh generation has been performed using the state-of-the-art pre- and post-processor MSC/PATRAN^[1]. The FE mesh of the model is shown in Fig. 2.

c. Boundary conditions and interfaces

The instrument is supported at four points. Therefore, constraints have been applied at the four nodes which are positioned at the location of the physical supports. Due to the complexity of the structure of the instrument and the lack of well-developed modelling techniques, the interfaces of the bolted joints, the sliding datum/bearings

and the spindle air bearings have been simplified, either as rigid connections or as MPC (multiple-point constraints). The contact area within the spherical joints has been modelled by using a few MPCs distributed over the annular contacting surface. Table 1 provides detailed information of the FE model.

d. The FE extraction of the natural frequencies and the mode shapes

After all pre-processing work has been completed, the FE model of the instrument was submitted to ABAQUS^[2]. Since only the first few vibration modes have dominant effects on measurements with the instrument, the first three vibration modes have been extracted. The extracted natural frequencies have been listed in Table 2 and the corresponding mode shapes are illustrated in Fig. 3, 4 and 5.

Experimental verification

To validate the developed FE model, an experimental modal analysis has been conducted. A vibration measuring system has been configured, as shown in Fig. 6, which consists of: a signal generator, an exciter, a power amplifier, an input transducer with a preamplifier, piezo-electric accelerometers with charge amplifiers, an analogue to digital card, a computer with data acquisition and the frequency analysis software, TestPoint^[3]. The experimental modal analysis technique^[4] has been used to obtain the necessary parameters in the experimental verification. The instrument was initially excited with a pseudo-random excitation. The response of the instrument was then measured and processed to obtain the transfer function, or frequency response function as shown in Fig. 10. From this, the natural frequencies of the instrument were determined and are listed in Table 2. The first peak shown in Fig. 10 is the natural frequency of the rubber supports of the instrument. The instrument was then excited with sinusoidal signals at the resonant frequencies that were determined previously. From the results of these measurements, the mode shapes of the instrument were extracted. The first three mode shapes of the instrument are depicted in Fig. 7, 8 and 9.

A comparison of the FE results with the measured results

A comparison of the three pairs of mode shapes, from the FE model and the instrument, (in Fig. 3 and 7, Fig. 4 and 8, and Fig. 5 and 9), shows a clear consistency between them. A comparison of the natural frequencies between the FE model and the instrument also shows a good agreement, as shown in Table 2. It is interesting to observe that all of the natural frequencies extracted from the FE model are lower than those from the test. This indicates that the stiffnesses of the connections used in the FE model are lower than those of the actual instrument. This underlines the importance of the interfaces, especially contact faces, to the successful FE modelling of metrology equipment.

Conclusions and Future work

An off-line FE model of the instrument has been created and verified by experiment. The work shows that it is possible to obtain a valid and meaningful FE model for the purpose of optimisations even before substantial progress has been made in the modelling of the interfaces. It is also revealed that the modelling of the interfaces is vital to the successful FE modelling of metrology equipment. To develop an on-line FE model of this instrument structure will require much further work to be done. The main work can be identified as the finite element modelling of: the spindle air bearings, the bolted joints, the sliding datum/bearings and the contact interfaces.

Acknowledgement

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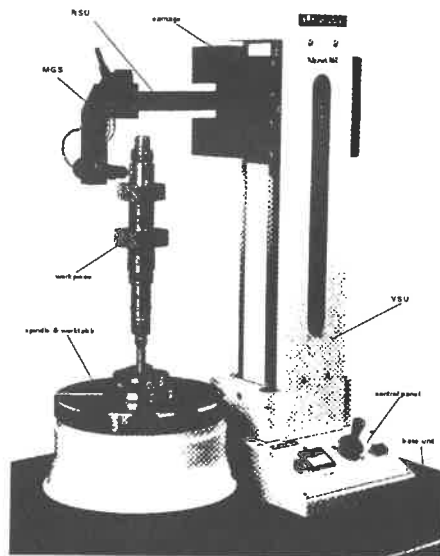


Fig. 1 Roundness measuring Instrument

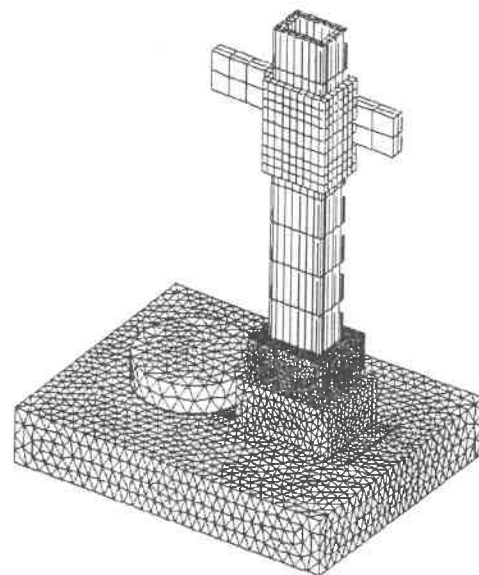


Fig. 2 FE model of the instrument

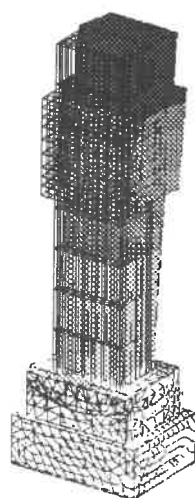


Fig. 3 First mode (FEM)

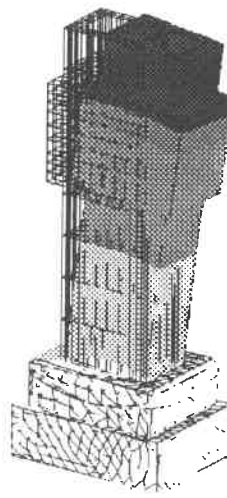


Fig. 4 Second mode (FEM)

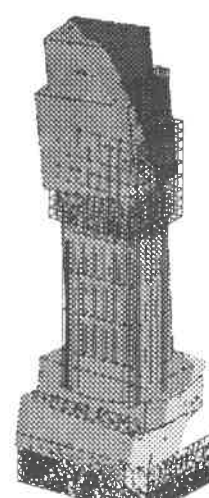


Fig. 5 Third mode (FEM)

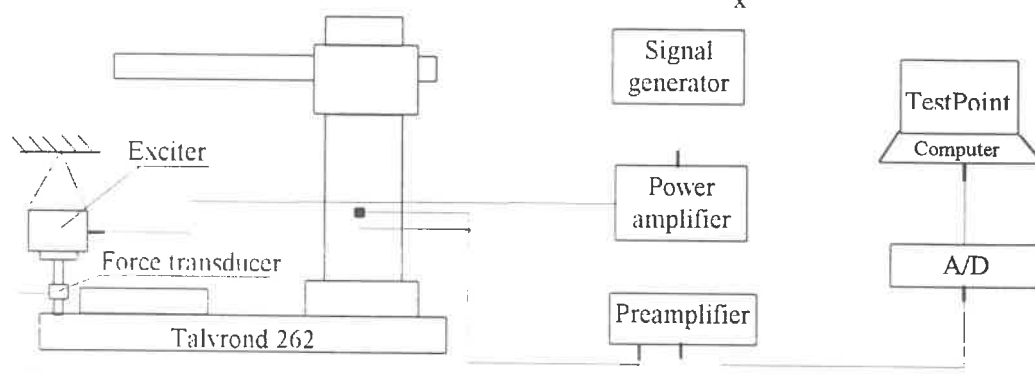


Fig. 6 Configuration of the vibration measurement system

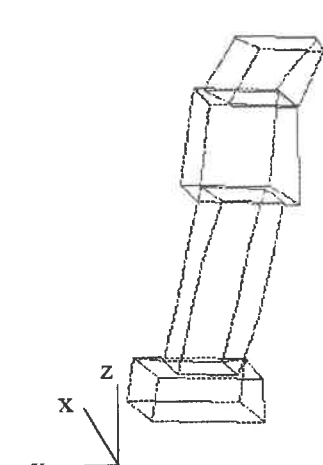


Fig. 7 First mode, $f=143$ Hz

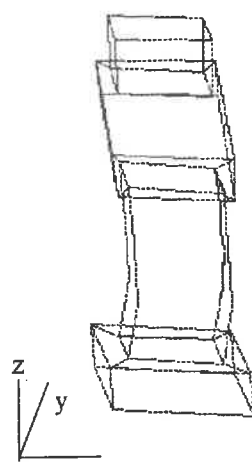


Fig. 8 Second model, $f=185$ Hz

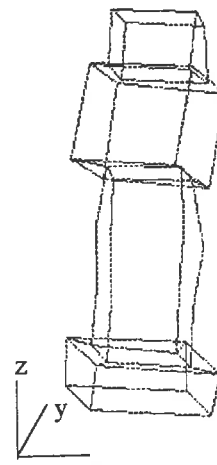


Fig. 9 Third mode, $f=355$ Hz

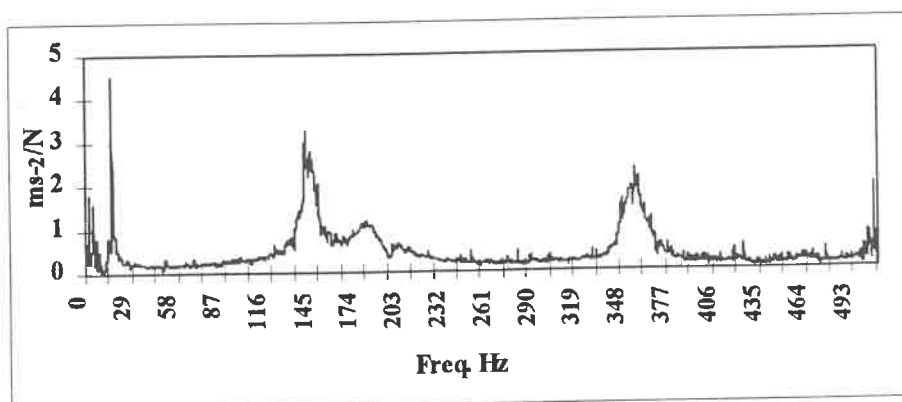


Fig. 10 Determination of the resonant frequencies

Table 1 Details of FE model of the instrument

	Elm type	No. of node	No. of elm	Material	E	d	λ
arm	C3D8	99	40	mixed	5.E10	4700	0.27
carriage	C3D8	560	252	Aluminium	6.9E10	2770	0.34
column	C3D8	702	416	Cast iron	1.65E11	7100	0.27
joint-up	C3D4	2105	9244	Cast iron	1.28E11	7250	0.27
joint-low	C3D4	2345	9593	Cast iron	1.28E11	7250	0.27
column base	C3D4	1019	3291	Cast iron	1.28E11	7250	0.27
insert	C3D4	2316	7401	Cast iron	1.28E11	7250	0.27
epoxy base	C3D4	4012	16542	Epoxy concrete	3.45E10	2269	0.2
spindle	C3D4	710	3012	Bronze	9.E10	8000	0.34
total		12976	49791				

Notes: E --- Young's Modulus, d --- density, λ --- Poison's ratio

Table 2 Comparison of the natural frequencies from the FE model and the instrument

	1st mode Hz	2nd mode Hz	3rd mode Hz
FEM	119.4	179.2	301.6
Experiment	143	185	355
difference	23.6	5.8	53.4
Error (%)	16.5	3.1	15
mode shape	bending	bending	torsional

MODEL FOR CMM TESTING BY LMS

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1. INTRODUCTION

CMMs are highly accurate, sophisticated metrological systems whose accuracy depends on large number of parameters. Their manufacturers are very much interested in producing the most accurate CMM with optimal production expenses.

Beside the verification of the CMM during installation process, it is necessary to check-up periodically its accuracy during exploitation. These activities are carried out by technical staff using measuring instrumentation from responsible institution (certified laboratory) and assisted by the users of equipment. These check-ups are necessary due to decreasing of the CMM's accuracy in exploitation process, caused by wear of slides, contamination of the air bearings, etc. These check-ups show if the values of the quality characteristics are within the limits stated by the manufacturer or the corresponding standard.

The parameters shown on Fig. 1. determine the quality of the CMM these are: (i) geometric accuracy of the axes, (ii) measurement uncertainty, (iii) kinematic accuracy, (iv) measurement of the etalon part and (v) accuracy of the software.

It is particularly emphasized that the elements of the classification of the quality parameters related to kinematic accuracy are taken as recommendations of OPTON who checks-up their newly manufactured CMM according to these recommendations. That means that they are advancing the standards who recommend accuracy check-up according to: (i) geometric accuracy of the axes, (ii) measurement uncertainty and (iii) measurement of the etalon part. This paper presents the results of research in measuring geometric accuracy of the axes and measurement uncertainty that are defined by national and international standards and rules by using self-developed IPM-CMM methodology [2, 3].

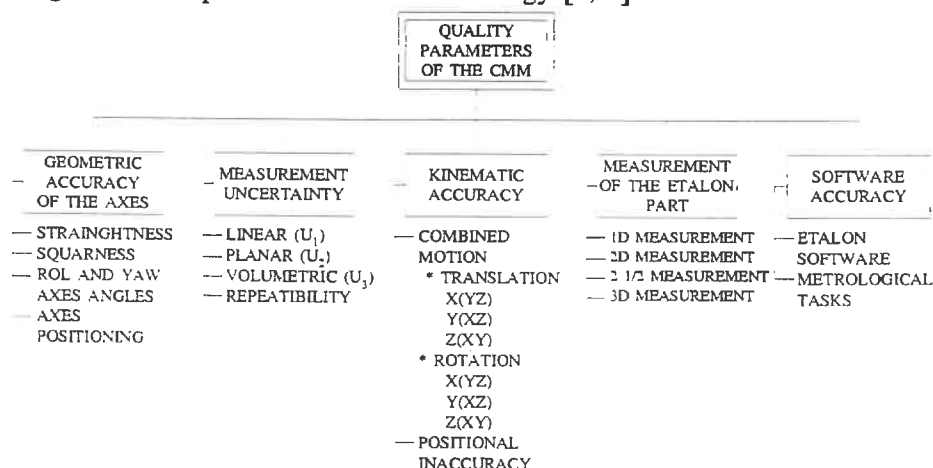


Figure 1. The parameters quality of the CMM

2. THE PROBLEM FORMULATION

Intensive development of the personal computers (PC), as well as wide availability of different accurate hardware accessories, caused the measuring systems based on PC computers to take a leading role over the expensive, specialized measuring equipment. Beside the lower price and exploitation expenses, the higher productivity, processing quality and processing accuracy as well as lower error rate is achieved.

The measurement systems, data processing and control based on PC computers are commonly called information systems for data acquisition; they are used for automation of the measurement systems in both laboratory and industrial environment. These information systems produce the high performance equipment with improved control characteristics and higher level of flexibility.

The developed model of information system for data acquisition, processing and check-up results analysis of the quality parameters of CMM consists of following parts: (i) laser measurement system (laser head and laser display), (ii) automatic compensator, (iii) digital parallel/serial interface, (iv) computer system and (v) software of the system.

Digital parallel/serial interface is built as a system for data acquisition using assembler for microcontroller 8051 and 2500 AD. The connection with the computer is established using RS 232 interface. The data acquisition is done by the software MEASURING written in programming language C++ 5.0. The basic form is shown on Fig. 2. This program creates and updates the file containing the measurement results. This file is later used by the main program for data processing, analysis and reasoning about the accuracy of the CMM that is coded in programming language Visual Basic 4.0.

3. THE SOFTWARE MODEL

The software support of the IPM-CMM system is developed under Windows operating system and the usage of this program follows the fundamental Windows principles. The users interface is based on windows and menu commands used in other Windows applications. After the installation of the program (during the installation the archive file will be automatically unpacked and the necessary directories will be created) was completed, one only needs to start the exe version of the program and the basic window shown on Fig. 3 will appear.

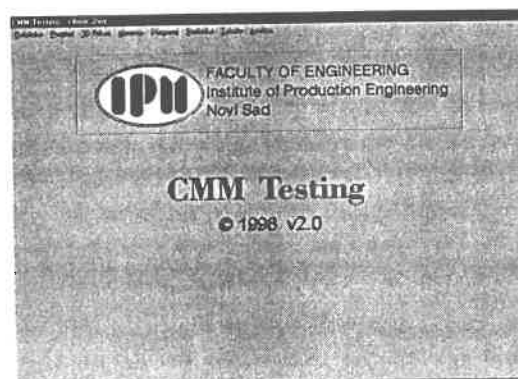
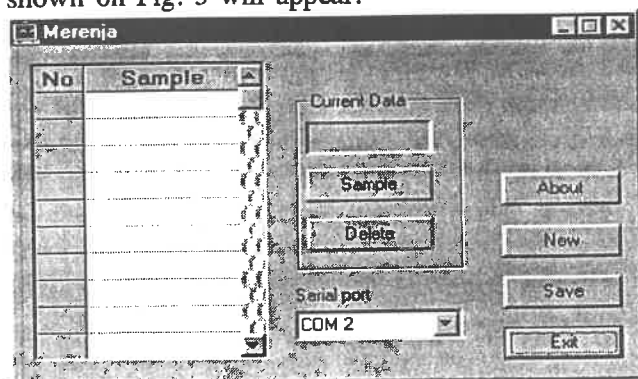


Figure 2. The screen of the software MEASURING Figure 3. The initial screen of the main software for CMM quality parameters testing

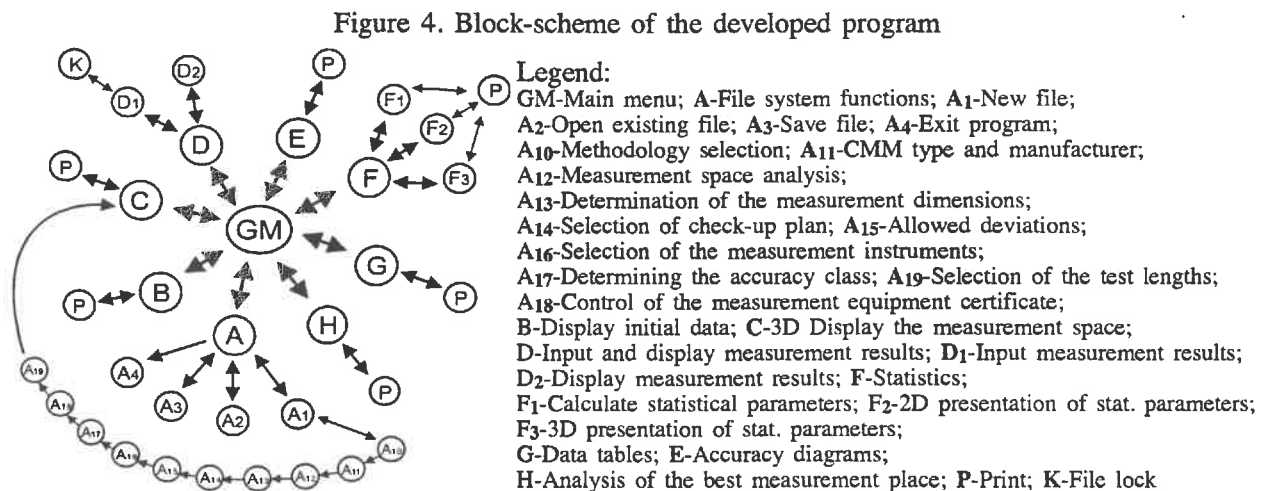
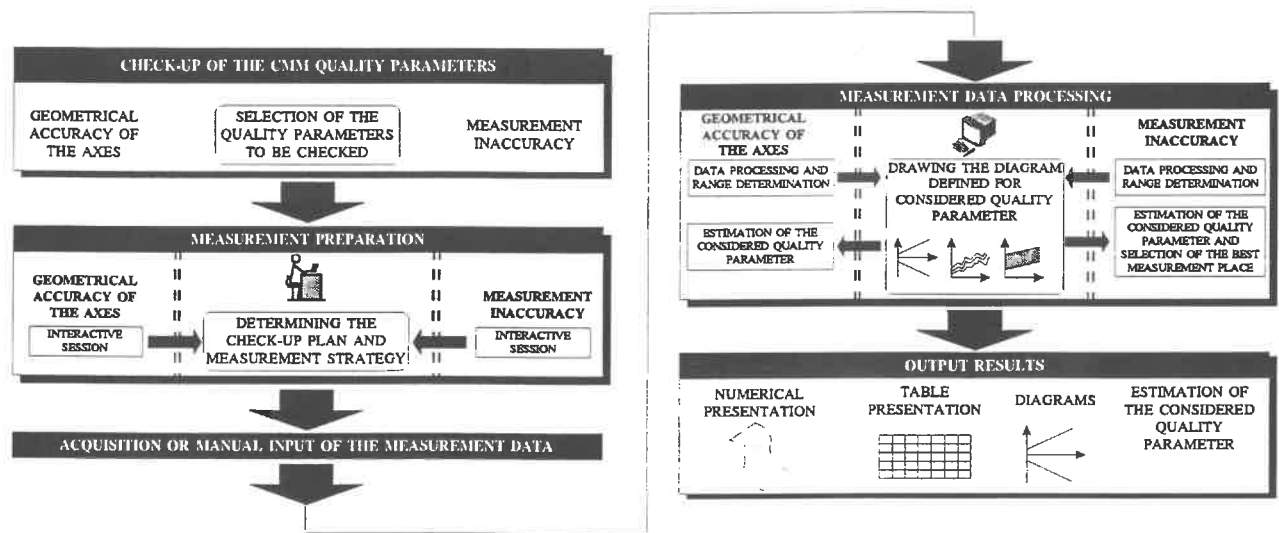
The process of the CMM quality parameters testing can be created in five (5) phases shown on the block-scheme of the developed program (Fig. 4.).

The Fig. 5. shows the state diagram of the system, where states are main menu and corresponding routines used for checking-up measurement uncertainty as quality parameter.

The results from particular segments are presented in the form of accuracy diagrams, tables, listings showing calculated percent of the measurement results out of range and 2D and 3D histograms (Fig. 6.).

4. CONCLUSIONS

The usage of the developed software integrated with the measurement process will be the most efficient if carried out by single operator. If the check-up is carried out by the team, the operator carrying out the measurement should be supported with the data about partition of the measurement space, values of the measurement-control lengths and other initial information that should be used for particular measurement in accordance with measurement lines that are supposed as a basis of the check-up methodology.



In the presented second version of the software system some options are in testing phase, but in order to develop the consistent software package the following modules should be developed: (i) extension of the software that will support quality parameters testing based on other national and international standards and methodologies, (ii) implementation of the working accuracy check-up based on measurement-control actions that supposes usage of the two-dimensional or three-dimensional measurement-control bodies, (iii) implementation of the step-gage as one-dimensional measurement-control bodies. The testing system designed and implemented to support all these functions could be a powerful tool for efficient and fast check-up of the CMM quality parameters. In the future, we are planning to develop the complete system based on the expert system approach.

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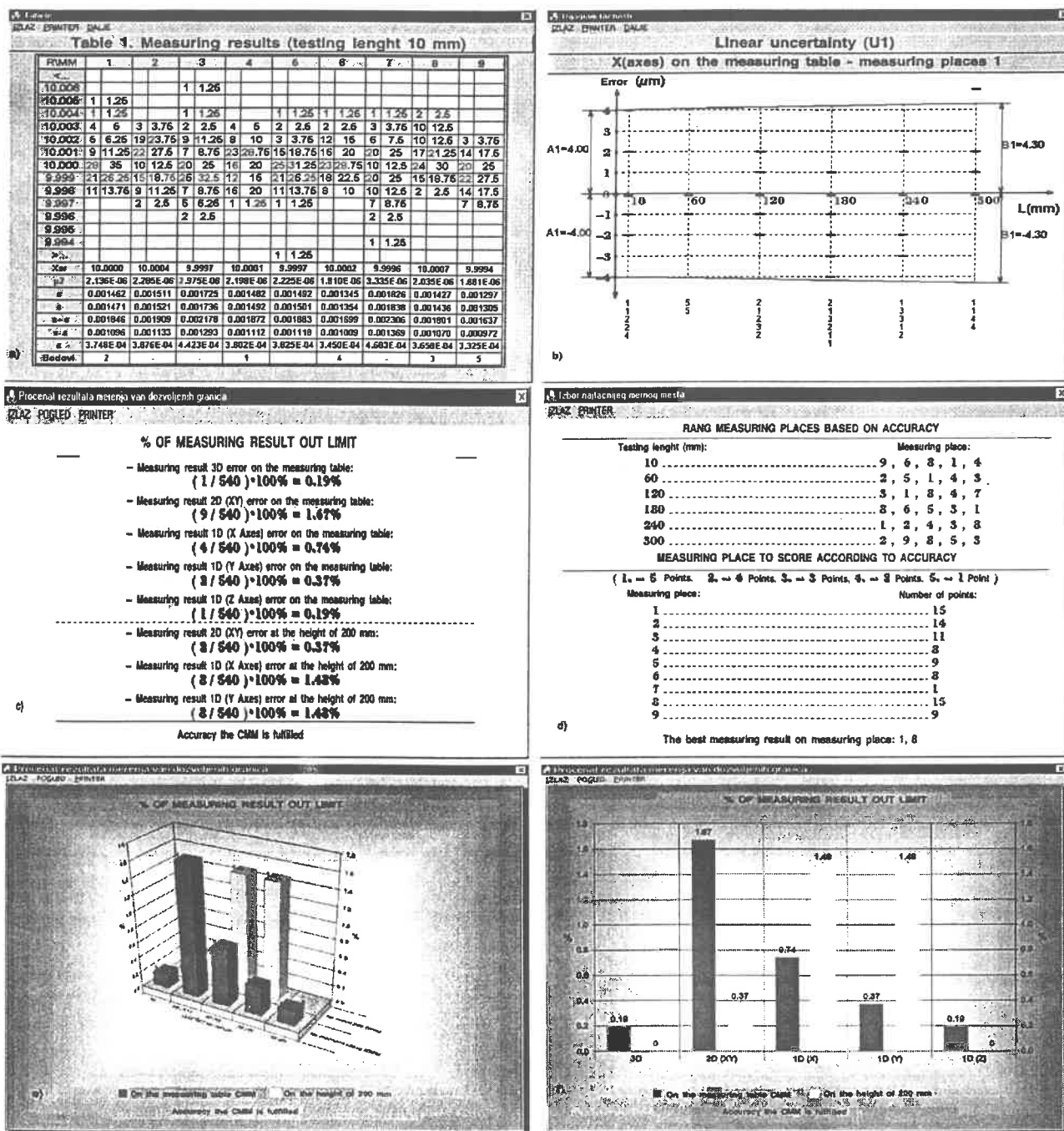


Figure 6. Some experimental results: a) measuring results (testing length 10 mm); b) linear uncertainty (U1); c) % of measuring results out limit; d) rang measuring places based on accuracy; e) % of measuring results out limit; f) some of e.

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A REDUCED SYNTHETIZATION MODEL FOR VIRTUAL CMM

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INTRODUCTION

The increasing flexibility and universality of coordinate measuring machines expanded its use in the manufacturing industry for the inspection of the dimensions of a large variety of products. They are faster than the conventional measuring instruments and actually they can substitute them in almost all measuring tasks, what increases the importance of knowing and improving their metrological performance and traceability of the measurements. In the last years, together with the search for more adequate calibration methods of CMM, a research trend that involves the mathematical simulation of the measurement possibilities of coordinate measuring machines arose as a possible solution to the problem of expressing the uncertainty of CMM measurements in concordance with the requirements of the new concept of traceability.

The actual calibration methods applied to CMM use different artifacts as reference to evaluate its metrological performance [1][5][6][7]. The measurement of such artifacts considers all machine inaccuracies, but it is traceable only for its specific measurement conditions. From some artifacts measurement it is possible to discriminate the individual errors of the machine for obtaining the error map, however this process is always very complex and in some cases incomplete.

The computer implementation of the mathematical model also called Virtual Coordinate Measuring Machine, combines and propagates the effects of all machine inaccuracies of a specific measurement task. The machine mathematical model is based on the principles of the rigid body kinematics and usually, the more used model requires the calibration of the 21 geometric errors of the CMM to obtain the machine volumetric error map [3],[4]. There are different forms of measuring the geometric errors [1], extensively mentioned in the literature as parametric calibrations. The error equations combine the effects of the parametric error in the preferential directions and allow to calculate the value of the volumetric error components at any coordinate point within the machine working volume.

On the other side, it is possible to measure directly the volumetric error components using the "grid" technique [3], where the volume is divided discretely by position measurements in the three preferential directions. The method doesn't require a model of the machine structure. This calibration technique is the more complete for diagnosing the machine behavior but it is more sensible to temperature influences because the time it requires to be realized. Frequently, the parametric error calibrations and the position measurements of the "grid" method are performed using a Laser Interferometer System, without considering the influences of the probe, but those influences can be foreseen and included in the machine model too. The error map of a Virtual Coordinate Measuring Machine can be used to compensate the machine systematic errors and the random part found can be combined to define the uncertainty for a specific measurement task.

The present study proposes a model to obtain the volumetric error map from the *partial* application of calibration space grid technique and the geometric analysis of the CMM structure behavior. The methodology was applied and a Virtual Moving Bridge CMM was developed. The results of error simulations were compared with those obtained for the same machine modeled and calibrated using the Synthesizing technique (measuring the 21 geometric errors).

MATHEMATICAL MODEL AND CALIBRATION

The calibration method associated to the model proposed in this work is characterized by the direct measurement of the volumetric error components (E_x , E_y , E_z) in three planes parallel to the preferential directions and will be called as "*partial grid*". The planes intercept each other at the center of the

working volume. The figure 1 shows the model principle. For each plane, the values of the error components corresponding to its two preferential directions are measured. Each interception line between the planes is parallel to a preferential direction and the error measured in those lines are always common to two planes.

For any point (X_i, Y_i, Z_i) placed in any of the 8 volumes defined by the interception of the three planes, the error in a particular direction is defined as a combination of the error variation between its projection points in two of the measured planes and in the interception line of them, see the figure 1b. Therefore, for any coordinate point (X_i, Y_i, Z_i) the volumetric error can be calculated using the following equations:

$$Ex_i - Ex_{XZ} = Ex_{XY} - Exc \quad \text{or} \quad Ex_i - Ex_{XY} = Ex_{XZ} - Exc \quad (1)$$

$$Ey_i - Ey_{XY} = Ey_{YZ} - Eyc \quad \text{or} \quad Ey_i - Ey_{YZ} = Ey_{XY} - Eyc \quad (2)$$

$$Ez_i - Ez_{XZ} = Ez_{YZ} - Ezc \quad \text{or} \quad Ez_i - Ez_{YZ} = Ez_{XZ} - Ezc \quad (3)$$

where Ex_i , Ey_i and Ez_i are the error at any point within the working volume, Ex_{plane} , Ey_{plane} and Ez_{plane} are the measured errors (with the partial grid) in the projection points of (X_i, Y_i, Z_i) in the planes XY, XZ and YZ and Exc , Eyc , Ezc are the measured errors at the projection point of (X_i, Y_i, Z_i) in the each interception line.

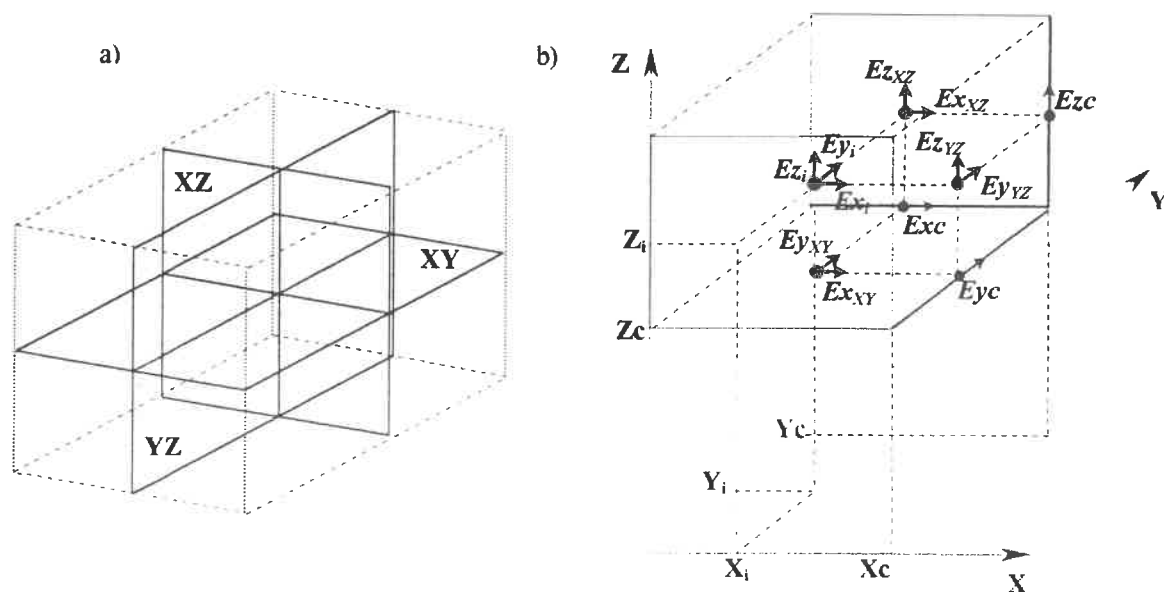


Figure 1: Partial Divided Volume: a) calibration planes.
b) reduced error synthetization model

Positioning measurements in the three preferential directions were made in three planes placed parallel to the machine axis and intercepting each other in the spatial point (150, 200, 125), see figure 1a. The measurements are 50 mm spaced in each direction, building three plane grids. They were made using a Displacement Laser Interferometer System. After correcting the initial point of each error curve measured, a machine error map was calculated using the expressions 3, 4 and 5.

Another error map was created using the error equations of a well tested synthetization model of the same machine [4]. The two maps were then extensively compared. The figures 2 and 3 show, as example some error curves. In figure 2 are presented the Ex error curves, measured with *partial grid* and synthesized for the same coordinate points in the grid measurement plane XZ.

The graphic of figure 3 contains the E_y error curves calculated with the partial grid model by different X position in a plane XY placed at $Z = 75$ mm and the error curves calculated for the same coordinate points with the synthesizing technique. The differences between the errors calculated for each point analyzed with the application of the two methods was in general, least than 10 % of the error value.

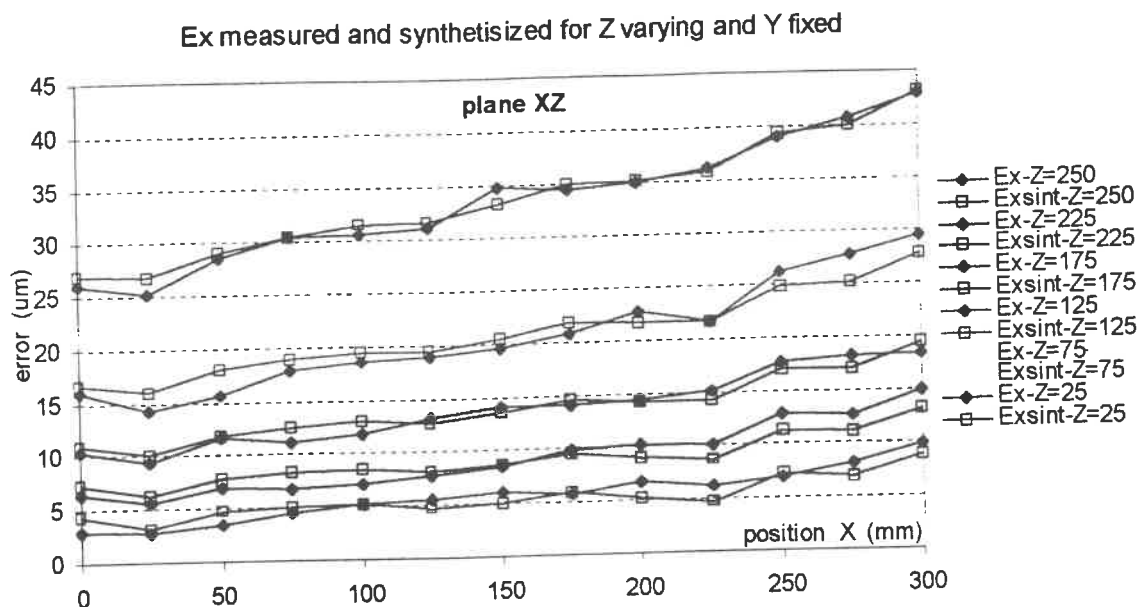


Figure 2: Errors in the direction X **measured** for different coordinate Z in the plane XZ of the partial grid calibration and calculated for the same points with the synthetization model

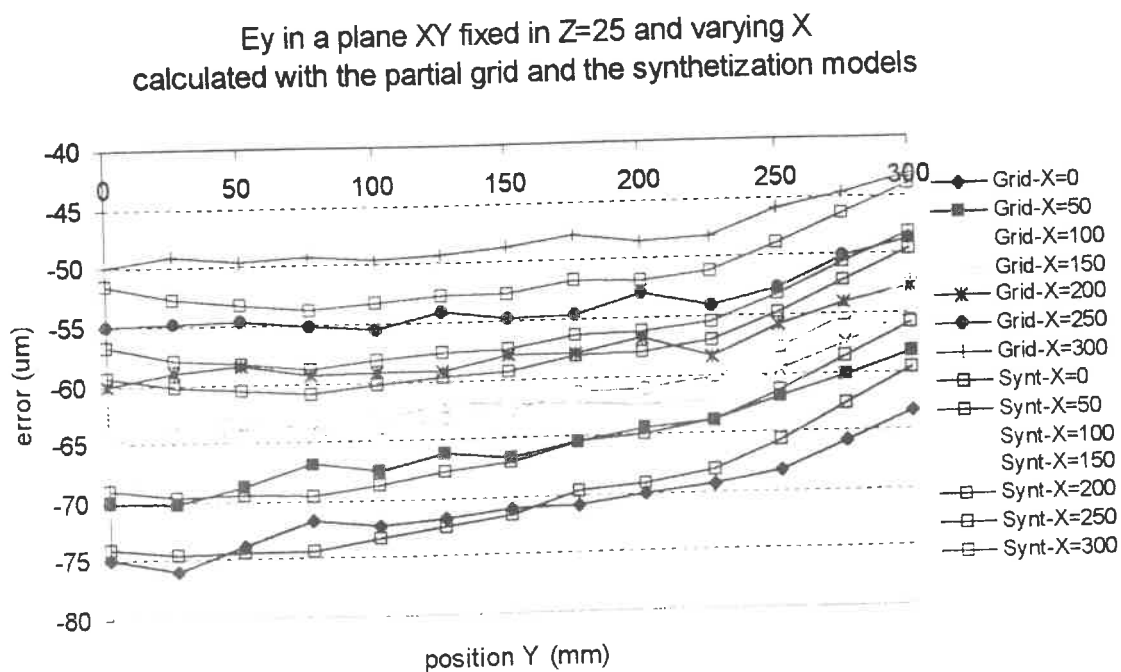


Figure 2: Errors in the direction Y **calculated**, for different coordinate X in the plane XY by $Z = 75$ mm. with the partial grid model and with the synthetization model

CONCLUSIONS

Considering the clear capability of traceability offered by the concept of a virtual coordinate measuring machine and searching for more representative uncertainty expressions, a model to calculate the volumetric error at any point in the machine volume was developed, using the partial application of grid techniques. A "moving bridge" CMM was virtually simulated with the proposed model and the results were compared with those obtained for the same machine modeled with the well known synthetization method. The excellent agreement found between the results shows the validity of the simple error equations proposed which carry least error propagation and permit an easier implementation.

Measuring position errors in only three planes diminished considerably the calibration time and it should still be better for automatic CMM. The calibration complexity was minimized since only one optic set was used and the method offers the possibility of measuring directly and eventually "on the fly" the error components. For CMM with error compensation system, the random contributions found for each point during the calibration can still be used to express the specific measurement uncertainty. Otherwise, the error map of the virtual machine can be used to compensate the systematic machine error.

Keywords: Virtual CMM, mathematical model, space grid techniques

Acknowledgment

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ARTEFACT PROBING FOR MACHINE TOOL AND PROCESS DIAGNOSIS

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Users of NC machine tools invest much time and effort in ensuring that the machines are constantly operating at their best, or at least that their performance remains stable over time. Their main concern is whether their expectations regarding the quality of parts produced are well founded. However with the more frequent use of machine tools with a touch trigger probe as measuring devices, the geometric integrity of the machine becomes even more important. This project was initiated to fulfill the need for a simple front line test that could be used frequently and rapidly to diagnose any change in the machine tool geometric status. The objective is not to replace methods such as laser interferometers, up-coming ball plate methods and telescopic ball bars. Instead, it aims at the early detection of the presence of faults with an ability to indicate potential causes, and make a judgement on their consequences on the process quality. In order to cause as little production time loss as possible, the method uses the already present touch trigger probe to take measurements on geometric features already present on the machine, or if that is not possible on a special low precision artefact mounted on the machine pallet.

The proposed method performs a three level diagnosis. Level 1 locates the faulty machine axes (e.g. X, Y, Z, A, B, C); level 2 suggests potentially faulty position dependent geometric parameters (e.g. scale, straightness, roll, tilt, etc.); level 3 makes a judgement on the ability to make or measure particular part features within a given tolerance.

The probing strategy aims at isolating one axis at a time. Ideally each test involves a main moving axis (the one under test) and small touch motions from the other linear axes. As a result, the artefact can be thought of as a collection of simpler sub-artefacts, each sub-artefact aiming at the diagnosis of a particular axis. Typically, linear step sub-artefacts are used for linear axes, and large step discs for rotary axes. Each step of each sub-artefact can be probed along the x, y and z directions along as much as possible of the range of motion of the targeted main moving axis. Two machines have been studied. The first one is an Omnimill 60 with a BXFZYA configuration (shown in Figure 1). Figure 2 shows the pallet mounted on this machine. The pallet has linear slots that can be used as linear step sub-artefacts for the X and Z axes (by rotating the B axis by 90 degrees). It also has radial slots that are used as a step discs to test the B axis. The second machine is a Mitsui Seiki with a CBXFZY configuration shown schematically in Figure 3. Since the probe cannot reach any portion of the pallet, a special artefact (Figure 4) was designed for this machine. All five axes can be tested. The artefact is modular. First, the full disc and linear step sub-artefact are mounted, and the C, X and Y axes tested. Then the half disc sub-artefact is assembled, and the B axis tested. Finally, a special step sub-artefact with varying heights is used for the Y axis. A special design was required in order to use the same stylus for all the tests.

The first time a test is performed, it provides a set of reference probing values. Ideally, these are acquired after a more complete assessment or overhaul of the machine (an alternative is to obtain measurements of the artefact departure from nominal using a coordinate measuring machines (CMM)). Future probing results provide a measure of the effect of variations of the machine geometric status (or of the absolute machine geometric parameters if CMM measurements were taken). The raw probing results, Ω_{act} , are the differences between the nominal and actual probed co-ordinates projected along the surface normal. Their value is influenced by the geometric imperfections of the artefact, the error in the location of the artefact on the machine, and by a number of machine axis geometric parameters (from moving and non-moving axes). The first step in the data processing aims at isolating the effect of moving axis parameters (Figure

5). The pre-processing subtracts an initial reference probing Ω_{ref} from the raw reading Ω_{act} , which gives a variational probing Ω_{var} . This removes the effect of artefact departure from nominal. The next step in the data processing (Figure 5) uses the knowledge of the probing strategy to remove the effect of quasi-non-moving axes as well as the effect of mounting errors of the artefact. This helps speed up the tests by requiring little or no set-up. It either removes the mean (for resultant scale tests) or the mean and slope of (for straightness tests), or the effect of step disc mounting error and axis location error from (for a rotary axis test), the variational probing.

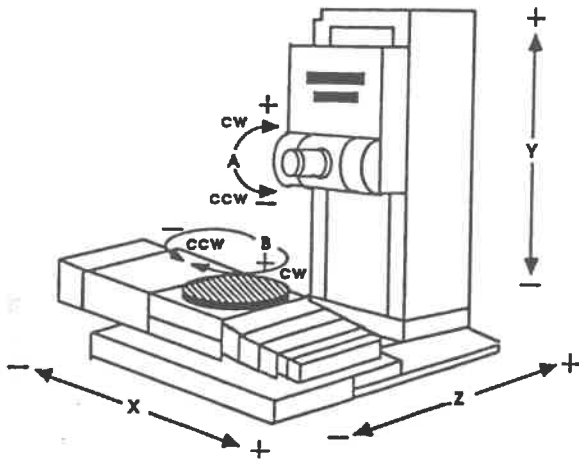


Figure 1 Omnimill 60 machining center.

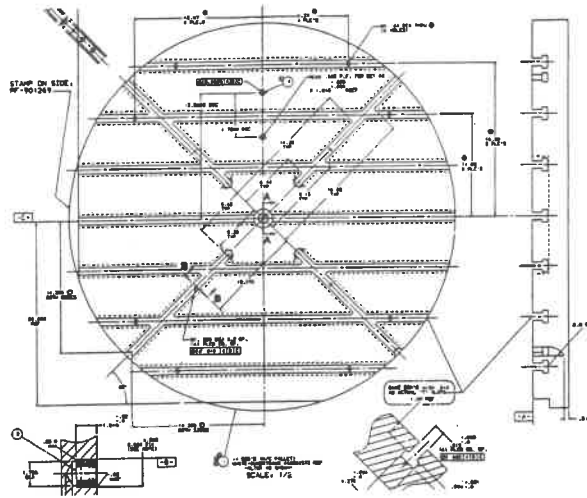


Figure 2 Omnimill 60 pallet.

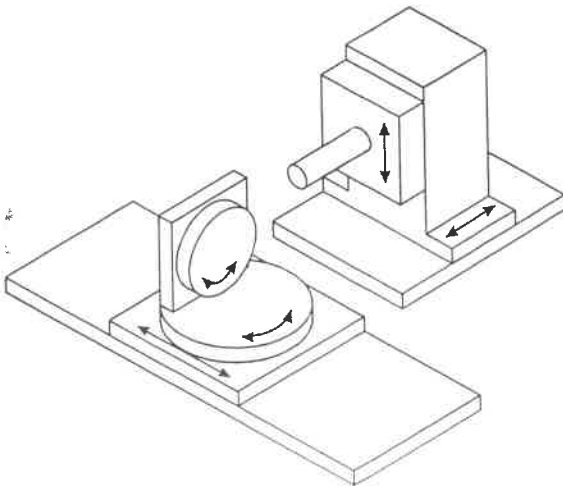


Figure 3 Mitsui Seiki machining center.

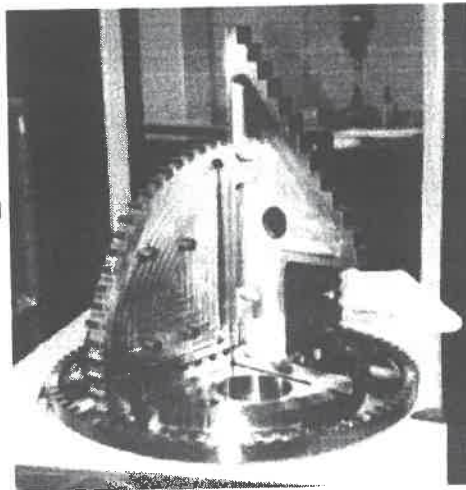


Figure 4 Artefact for Mitsui Seiki.

Once processed, the data obtained represents three sets of resultant errors per axis. For a linear axis, these resultant errors are in the x, y and z directions and can be assimilated to a resultant scale and two resultant straightness tests (e.g. $\Omega_x(x)$, $\Omega_y(x)$ and $\Omega_z(x)$ for the X axis). For rotary axes, the resultant errors can be assimilated to tangential, radial and axial resultant errors. The level 1 diagnosis consists in a visual inspection of these resultant errors in graphical format as a function of the main moving axis position. Some threshold values can be superimposed on the graph to assist the operator decide on the pursuance or otherwise of production. A squareness error between linear axes is detected using measurements on the

outside diameter of the step disc by fitting the best ellipse to these points. If level 1 results are within acceptable limits then the analysis can stop here. Otherwise, we proceed to level 2.

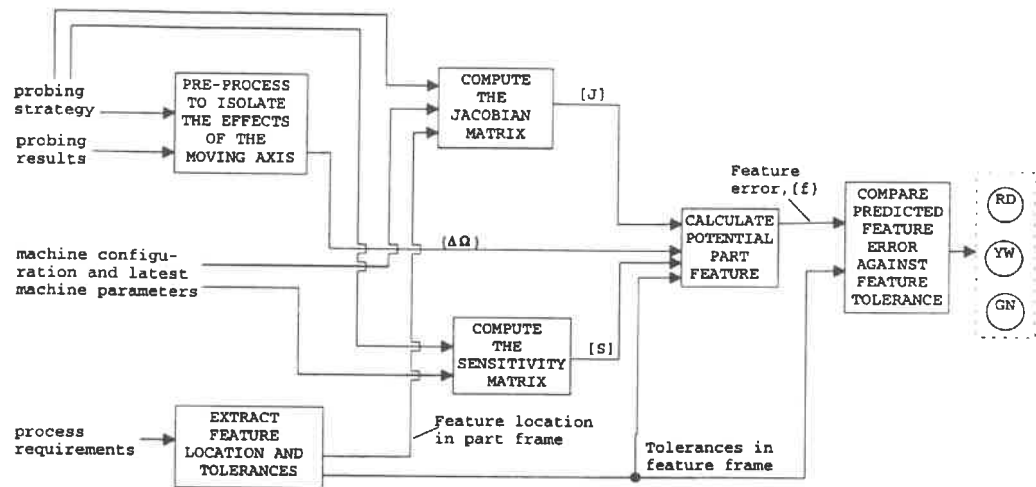


Figure 5 Diagram of the machine and process diagnosis tool concept.

Once the resultant variational errors due to the moving axis have been isolated, their potential causes at the axis level can be evaluated from the knowledge of the probing location, its direction, and from the knowledge of the machine kinematics. This can be done by analysing the S matrix (it is a Jacobian matrix) relating the machine geometric parameter vector $\{q\}$ to the probing error vector $\{\Delta\Omega\}$.

$$\{\Delta\Omega\} = S\{q\} \quad (1)$$

The S matrix is produced using a powerful matrix formulation of the Newton-Euler equations. This formulation produces an analytical Jacobian matrix from the 6×1 twists describing the machine kinematics. The S matrix is analyzed to detect significant matrix elements (i.e. over a pre-determined threshold) in order to conclude on the causal relationship between a test and a geometric parameter. The machine operator, following a visual inspection of the three tests pertaining to a particular axis, selects a triplet representing the tests which are affected (e.g. 0,1,0 or 0,1,1 etc.). Each binary digit of the triplet indicates whether (1) or not (0) the corresponding test for a particular axis shows an excessive variation over its range of movement. On the basis of the triplet, the program suggests which of the six position dependent geometric parameters is or are potentially at fault.

Table 1 shows which parameters affect each probing test in the case of the OM60 machining center X axis. Table 2 indicates on each row, a possible combination of tests showing degrading resultant errors. This combination is shown in the first three columns, followed by the potentially faulty parameters in the following columns.

Table 1 Causal relation between geometric parameters and resultant errors.

	$e_x(x)$	$e_y(x)$	$e_z(x)$	$\gamma_x(x)$	$\gamma_y(x)$	$\gamma_z(x)$
$\Omega_x(x)$	✓	0	0	0	✓	✓
$\Omega_y(x)$	0	✓	0	✓	0	✓
$\Omega_z(x)$	0	0	✓	✓	✓	0

Table 2 Level 2 diagnosis for the X axis

Affected probing tests			Potentially faulty parameters of that axis					
$\Omega_x(x)$	$\Omega_y(x)$	$\Omega_z(x)$	$e_x(x)$	$e_y(x)$	$e_z(x)$	$\gamma_x(x)$	$\gamma_y(x)$	$\gamma_z(x)$
✓	0	0	✓	0	0	0	0	0
0	✓	0	0	✓	0	0	0	0
0	0	✓	0	0	✓	0	0	0
✓	✓	0	✓	✓	0	0	0	✓
✓	0	✓	✓	0	✓	0	✓	0
0	✓	✓	0	✓	✓	✓	0	0
✓	✓	✓	✓	✓	✓	✓	✓	✓
0=no, ✓=yes								

The level 3 diagnosis uses the sensitivity matrix, S , and a similar Jacobian matrix, J , translating the machine errors into part feature errors. Basically, it generates a machine that would have produced the obtained probing results, and predicts how this machine would fair at the part features. The prediction is compared to the drawing tolerance. A process requirement file is generated on the basis of the manufacturing process plan. The process plan contains detailed information on the type, location, size and tolerance of all part features to be machined. An alternative to a diagnosis based on the process is a volumetric diagnosis. In this case, the part feature error vector is produced using uniformly distributed positions within the machine workspace. This has the advantage of producing a less process dependent assessment, and so more machine orientated and of more general use.

This information is used to compute the appropriate Jacobian matrix, here called J , which describes the impact of the machine axis geometric parameter vector $\{q\}$ on the part feature error vector $\{f\}$ as follows:

$$\{f\} = J\{q\} \quad (2)$$

The final step is to solve an equation similar to Eq. 1 for $\{q\}$, then substitute in Eq. 2 to conclude on $\{f\}$, and compare it with a part feature tolerance vector $\{t\}$. This requires the definition of probing test image points, which are probing test locations for which the main moving axis is at the same location as for the part feature. So, $\{\Delta\Omega\}$ in Eq. 1 must be interpolated since a probing point most probably is not available at that exact location. Similarly the S matrix is developed at that image point. Even so, a unique solution does not exist for $\{q\}$ since the number of unknown exceeds the number of equations because of the structure of the data gathered. A number of schemes are thus under study to generate representative solutions. One scheme uses the pseudo-inverse of Moore-Penrose [1] of S , S^+ , and provides a minimum norm solution for the parameters resulting in the following system:

$$\{f\} = J S^+ \{\Delta\Omega\} \quad (1)$$

Experimental measurements and simulations have been made on the Mitsui Seiki showing the successful extraction of the resultant errors caused by the moving axis for the level 1 processing both for linear and rotary axes. Simulations conducted on a computer model of the OM60 show that level 3 assessment of whether a feature will be within tolerance or not is successful in about 60 to 100% of cases depending on the machine degradation. The proposed method and associated program will be further tested in an industrial environment: this next phase will provide extensive experimental data and shop floor personnel feedback to fully validate the method.

DISTRIBUTION-FREE STATISTICAL METHODS FOR ANALYZING LINEAR TOLERANCE ACCUMULATION

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Statistical methods for specifying part dimensions have been used in industry for over forty years [Brooks 56], and work to formalize these methods has begun recently in international (ISO) dimensioning and tolerancing standards. The primary motivation for statistical tolerancing (ST) is cost reduction through the use of less restrictive tolerances. One method of ST specification is to specify acceptable values for the mean (μ) and variance (σ^2) of the statistical distribution associated with a population of part dimensions. This can be done by defining regions in a (μ , σ) distribution-parameter space [Srinivasan 94], denoted STZones. The probability that the dimension of a randomly selected part will not meet specified worst-case limits is called the *failure rate* of the population.

Failure rate estimates based on the assumption of normal statistics often under-estimate drastically the actual failure rate if the population statistics are non-normal [Braun 97]. Thus failure rate criteria are very sensitive to the type of distribution, and lead one to seek criteria which are independent of the distribution, i.e. are distribution free (DF). The most obvious and straightforward DF methods are based on the Chebyshev inequality. The bounds produced by Chebyshev arguments are very conservative, and this paper describes methods for improving (tightening) them considerably.

This paper describes two levels of improvement over the Chebyshev bounds for distribution-free inference of assembly failure rate. The first level utilizes the one-sidedness of the assembly criterion (non-negative gap size) to sharpen the Chebyshev bound. Considerably greater improvements can be attained by recognizing that assemblies are collections of parts, each of which has (presumably) met a DF criterion. This fact – that several / many DF criteria have been met – is analogous to numerical averaging in distribution-dependent dimension chains; it is reflected mathematically in DF criteria, as points on the boundary of the gap STZone (constructed through Minkowski summation – a decidedly new method in tolerancing applications [Srinivasan 97a]). These boundary points *must* be the result of component distribution parameters which are each on the boundary of their respective STZones. By identifying the population parameters for the component populations, we can infer the limiting distribution of the gap population – that distribution which results in the highest assembly failure rate.

Direct Inference

A critical clearance dimension, or *gap size*, is the value of interest in linear, or one-dimensional, assembly analysis. The gap size is a linear combination of component part dimensions (a dimension chain), and the criterion for assembly is that the gap size be non-negative. Note that the gap size criterion is one-sided – it is bounded only from below (by zero.)

Consider a population of assemblies for which the population parameters μ and σ of the critical 'gap' in the assembly are known, or can be inferred from the component population parameters. The question we pose is "*what is the probability that a selection of components will fail to assemble, i.e. the gap size is less than zero?*"

For the remainder of this document we will use x_i to denote a random variable corresponding to a component size, and y to denote a random variable corresponding to the gap, where y is a linear combination of the x_i 's.

Chebyshev

The Chebyshev inequality bounds the probability that a sample is at least $k\sigma$ away from the mean, or

$$P[|y - \mu| \geq k\sigma] \leq \frac{1}{k^2}, k > 0. \quad (1)$$

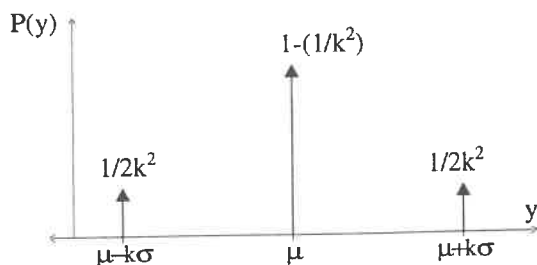
However, for non-interference at the gap, we are concerned only with the probability that the sample is $k\sigma$ less than the mean, or

$$P[y \leq \mu - k\sigma], \quad (2)$$

which is (as pointed out in [Scholz 97])

$$\leq P[(y \leq \mu - k\sigma) \vee (y \geq \mu + k\sigma)] = P[|y - \mu| \geq k\sigma] \leq \frac{1}{k^2}. \quad (3)$$

The distribution shown in Figure 1 satisfies the equality case of Equation (1). However, we observe that the actual number of failures (when $y < \mu - k\sigma = 0$ for our assembly examples) in this limiting distribution is only one half of the $1/k^2$ value, or $1/2k^2$.



Note: The notation $1/2k^2$ next to the leftmost arrow is shorthand for

$$\frac{1}{2k^2} \delta(y - (\mu - k\sigma)),$$

where δ represents a generalized delta function. The other delta functions have similar notation.

Figure 1: Limiting distribution exhibiting Chebyshev equality.

The question we now ask is "while Figure 1 shows an assembly failure rate of $1/2k^2$ (where k is determined by $\mu - k\sigma = 0$), are there distributions which have higher failure rates, and can the $1/k^2$ limit be reached?"

The one-sided limit

If we are concerned only with populations (with specified μ and σ) which have the greatest one-sided failure rate, we seek the maximum probability for Equation (2). The limiting distribution is the one shown in Figure 2.

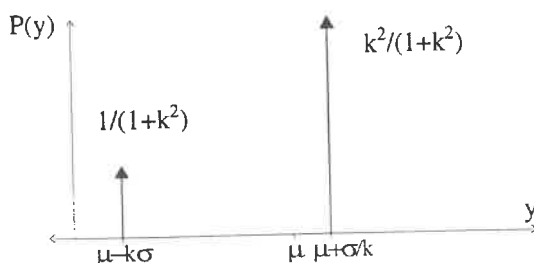


Figure 2: Limiting distribution for one-sided failure rate.

So, for one-sided bounds on a distribution, we can say

$$P[y \leq \mu - k\sigma] \leq \frac{1}{1+k^2}, \quad k > 0. \quad (4)$$

note: For the two cases (Chebyshev and one-sided limits) discussed above, we have allowed generalized (delta) functions at the failure limit ($y=0$) to be considered failures. This is acceptable if we choose our delta function to be the limiting case of, say, an exponential distribution with the tail in the negative direction.

Extension of Inference (composition)

The previous section discussed direct inference, which is appropriate for any population with known (μ, σ) values. Our second level of inference utilizes not only the gap population parameter values, but the fact that the gap

population is composed of component populations, which in turn have bounded (or known) μ , σ parameter values. The example which follows shows how this information may be exploited to bound the assembly failure rate at a lower value than allowed by the direct inference techniques.

Example

Consider the simple assembly shown in Figure 3. In this case there are only two component dimensions which contribute to the resultant gap dimension. We say an assembly instance (the random selection of a pair of components) is a failure if the gap dimension ($y = x_1 - x_2$) is negative.

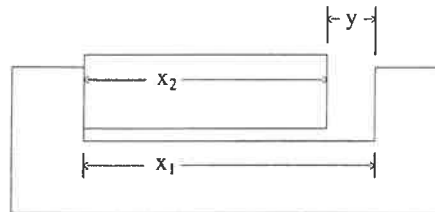


Figure 3: A simple assembly example.

The component population parameters for x_1 and x_2 are bounded as shown in Table 1 and Figure 4. The population parameters possible for gap population may be easily calculated [Srinivasan 97a], [Srinivasan 97b].

	μ	σ
x_1	[10.0, 10.5]	[0, 0.212]
x_2	[8.5, 9.0]	[0, 0.212]
y (calculated)	[1.0, 2.0]	[0, 0.300]

Table 1: Population parameters for simple assembly.

Now consider point A in Figure 4. This point is of interest because it represents populations with the highest potential assembly failure rate, corresponding to the largest value of k ($k=(\mu-0)/\sigma$). If we make no assumptions about the nature of the distribution of y , we may use the Chebyshev inequality or the one-sided inequality to place an upper bound on the failure rate for populations which have the (μ, σ) values of (1.0, 0.3). The k value for the gap population is 3.333, and the Chebyshev inequality (Equation 1) bounds the failures at no more than 9.00% while the one-sided bound (Equation 3) is 8.26%.

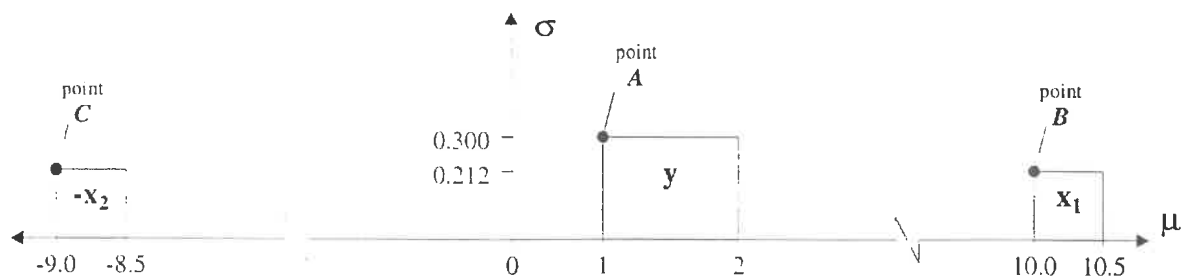


Figure 4: (μ, σ) parameter bounds on x_1 , x_2 , and y populations.

In order for the gap population to have the parameter values at point A, the population parameters for component sizes x_1 and x_2 must have the values at points B and C respectively. For the rectangular parameter region of y , the 'corner' points of the region are associated with a unique combination of component parameters.

We now want to find how the x_1 and x_2 distributions may contribute to the y distribution so that a maximal failure rate is attained. Observe that while the x_1 and x_2 (μ, σ) parameters are known (points B and C), the actual distributions are not specified. Let's first use the one-sided analysis to find the assembly failure rate if each

component distribution contributes a maximal fraction of its population at 0.5 units from the population mean. (i.e. there are some x_1 parts and some x_2 parts with size 9.5) In this case, the component and gap distributions would be as shown in Figure 5, and the assembly failure rate $\cong 2.3\%$.

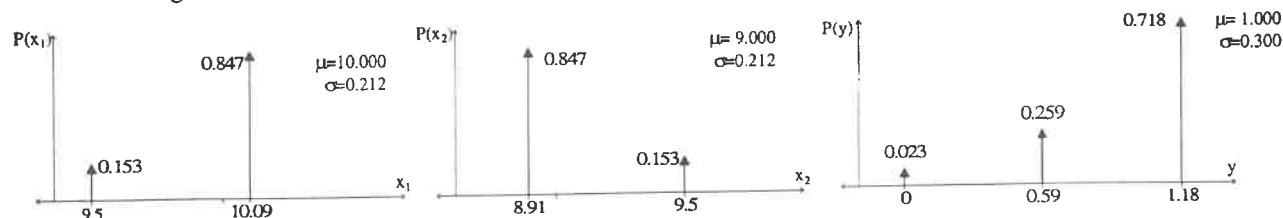


Figure 5: Possible component distributions contributing to an assembly failure rate of 2.3%.

In order to find a bound for the assembly failure rate, we parameterize the maximal assembly failure rate in terms of the relative contribution of the x_1 , x_2 distributions as shown in Figure 6. In order to have any failures, the x_1 population must have some parts which are at α less than the mean of 10, while the x_2 population must have some parts at the same size: $(1-\alpha)$ greater than the mean of 9. If we select one each of these parts, we will have an assembly which "just fails". We now compute the failure rate in terms of α .

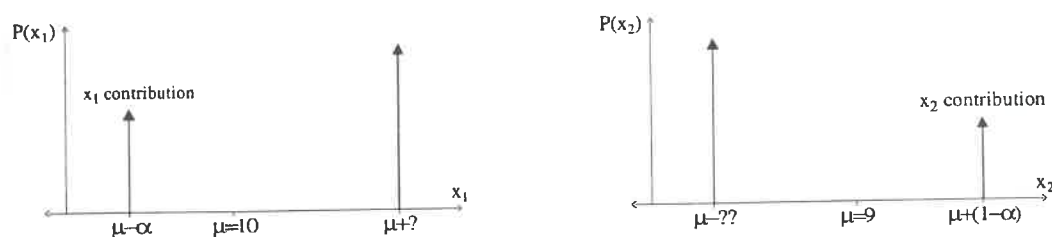


Figure 6: Component distributions showing contribution variable α .

Figure 7 shows the maximum possible assembly failure rate as a function of the relative contribution α . Since the population parameter regions for x_1 and x_2 have the same size and shape, it is not surprising that the plot is symmetric. It is surprising that a local minimum value occurs when each component contributes equally, which corresponds to the case shown in Figure 5.

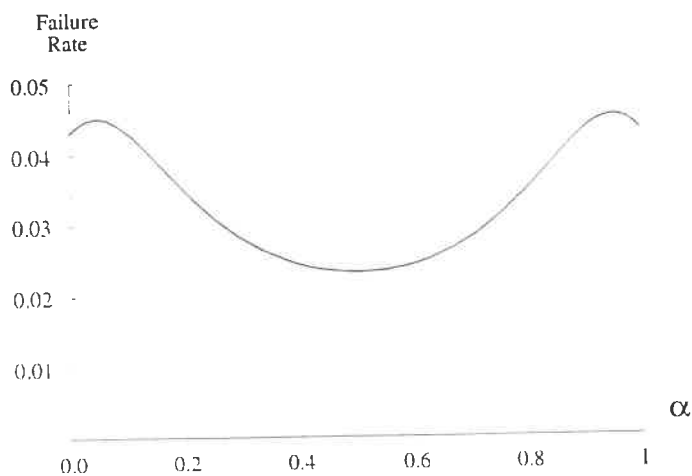


Figure 7: Assembly failure rates for different component contributions (α).

If we choose one of the maxima from Figure 7 ($\alpha = 0.0472$) to form the component distributions, we obtain the component and gap distributions shown in Figure 8. This case gives a (maximum) failure rate of 4.5%.

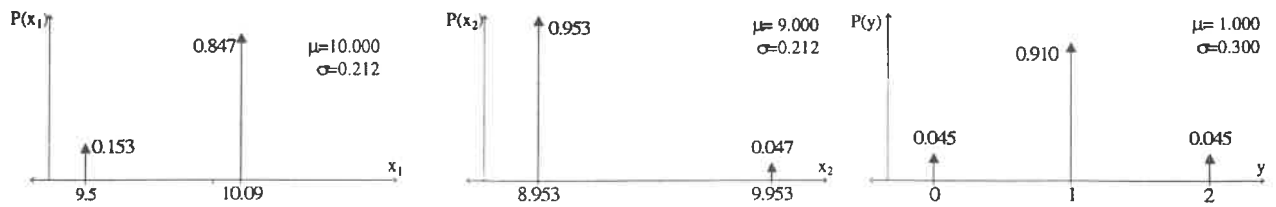


Figure 8: Distributions resulting in maximum possible failure rate for 2-part example.

Summary:

We have shown that the use of the mean and variance data of component distribution parameters results in tightening the DF bounds on the assembly failure rate. In our simple two-part example, we reduced the upper bound on our failure rate by one half (from 9.0% to 4.5%.) This composition technique may be extended to assemblies with more parts and also to population constraints that are not strictly DF, such as assuming the populations of component dimensions (the x_i 's) have unimodal distributions.

In the full paper, we study examples with more than two parts, including a five-part example taken from [Srinivasan 97b]. The improvement over the Chebyshev DF bound is increased further as the number of parts in the assembly grows. Uniqueness considerations for points in the gap STZone are also addressed.

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Analysis error depending on amount of shear in lateral shearing interferometry

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Introduction

An LS interferometer is one kind of common-path interferometer, so it is little affected by mechanical vibration and air turbulence. It consists of a few number of optical components. In the interferometer, the interference pattern represents the difference between a wavefront and the sheared wavefront. Many types of LS interferometers have been developed. In their interferometers, there is an LS interferometer with optical parallel¹. On this type, reflected wavefronts on front and back of the optical parallel are interfered. We have developed an LS interferometer of this type with zone plates. And a spherical mirrors and aspherical mirrors have been measured by the interferometer.

To reconstruct the shape under test from the LS interferogram, many methods³⁻⁵ of reconstruction have been suggested. In a method⁶ suggested by Rimmer and Wyant, the wavefronts are reconstructed with the polynomials and fitting. In other methods, there is a method with integration of phase distribution. In this method, process of analysis is easy, and the wavefront is reconstructed quickly. Yatagai et al. used this method in a measurement⁷ of a mirror. When the amount of shear is small, accurate shape is reconstructed. When the amount of the shear is large, an error included in reconstructed shape is large. To measure wavefront under test accurately, error in

reconstructed shape must be compensated. In this paper, a method of compensation for the error with integration is suggested.

Calculation of the error

First, to investigate the error with integration, in a measurement of a shape expressed by Zernike polynomials $U_i(x,y)$ with LS interferometer, a shape to be reconstructed with integration are calculated. i is a number of Zernike polynomials. Zernike polynomials are useful in various ways, for example, it is used for fitting or for a shape under test in a simulation. Fig. 1 shows a schematic diagram of LS interferences.

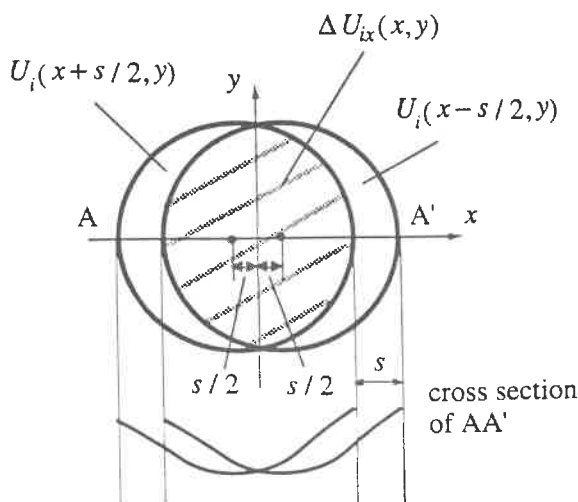


Fig.1 Schematic diagram of an original wavefront and the sheared wavefront.

In a shape $U_i(x, y)$ under test, a difference $\Delta U_{ix}(x, y)$ between the shape under test and the shape sheared with s of the amount of shear is given by

$$\Delta U_{ix}(x, y) = U_i(x + s/2, y) - U_i(x - s/2, y). \quad (1)$$

After each side of equation (1) are divided by s , a shape $U_{ix}(x, y)$ is reconstructed with integrating the equation in x -direction. $U_{ix}(x, y)$ is given by

$$U_{ix}(x, y) = \int \Delta U_{ix}(x, y) / s \, dx. \quad (2)$$

In the shape $U_{ix}(x, y)$, the information of y -direction cannot be reconstructed. So $U_i(x, y)$ is sheared in y -direction too, $\Delta U_{iy}(x, y)$ is calculated. $\Delta U_{iy}(x, y)$ is given by

$$\Delta U_{iy}(x, y) = U_i(x, y + s/2) - U_i(x, y - s/2). \quad (3)$$

After each side of equation (3) are divided by s , a shape $U_{iy}(x, y)$ is reconstructed with integrating the equation (3) in y -direction. $U_{iy}(x, y)$ is given by equation (4).

$$U_{iy}(x, y) = \int \Delta U_{iy}(x, y) / s \, dy. \quad (4)$$

By given one line of $U_{iy}(x, y)$ to $U_{ix}(x, y)$, the shape of wavefront $U_i(x, y)$ is reconstructed. In each Zernike polynomials, an equation of the shape analyzed with integration is calculated. Zernike

Table 1. Zernike polynomials and The polynomials of the reconstructed shape

i	$U_i(x, y)$	$U_i'(x, y)$
1	1	-
2	x	x
3	y	y
4	$2xy$	$2xy$
5	$-1 + 2y^2 + 2x^2$	$2y^2 + 2x^2$
6	$y^2 - x^2$	$y^2 - x^2$
7	$3xy^2 - x^3$	$3xy^2 - x^3 - s^2x/4$
8	$-2x + 3xy^2 + 3x^3$	$-2x + 3xy^2 + 3x^3 + 3s^2x/4$
9	$-2y + 3y^3 + 3x^2y$	$-2y + 3y^3 + 3x^2y + 3s^2y/4$
10	$y^3 - 3x^2y$	$y^3 - 3x^2y + s^2y/4$
11	$4y^3x - 4x^3y$	$4y^3x - 4x^3y - s^2xy$
12	$-6xy + 8y^3x + 8x^3y$	$-6xy + 8y^3x + 8x^3y + 2s^2xy$
13	$1 - 6y^2 - 6x^2 + 6y^4 + 12x^2y^2 + 6x^4$	$-6y^2 - 6x^2 + 6y^4 + 12x^2y^2 + 6x^4 + 3s^2x^2 + 3s^2y^2$
14	$-3y^2 + 3x^2 + 4y^4 - 4x^4$	$-3y^2 + 3x^2 + 4y^4 - 4x^4 - 2s^2x^2 + 2s^2y^2$
15	$y^4 - 6x^2y^2 + x^4$	$y^4 - 6x^2y^2 + x^4 + s^2x^2/2 + s^2y^2/2$

polynomials and the equation obtained by integration up to 15 term are shown in table 1. The polynomials of the shape reconstructed with integration are consists of components of Zernike polynomials and polynomials of components of errors caused by the amount of shear. However a term without a parameter x and y , for example 1 included in the first term of Zernike polynomials, cannot be reconstructed. Polynomials of the error $E_i(x, y, s)$ up to 15 term are shown in table 2. From the polynomials obtained by the calculation, it is found that the shape $U'_i(x, y)$ analyzed by integration is given by

$$U'_i(x, y) = U_i(x, y) + E_i(x, y, s). \quad (5)$$

$E_i(x, y, s)$ is polynomials of the error.

A method of compensation for the error caused by integration

If a wavefront $W(x, y)$ is continuous and sufficiently smooth, the wavefront can be described by Zernike polynomials $U_i(x, y)$. The wavefront is given by

$$W(x, y) = \sum B_i U_i(x, y). \quad (6)$$

where B_i are coefficients of polynomials. When an LS interferogram of the wavefront is analyzed by integration, the analyzed shape $W'(x, y)$ is given by

$$W'(x, y) = \sum B_i U'_i(x, y). \quad (7)$$

By substituting equation (5) for equation (7), equation (8) is given.

$$W'(x, y) = \sum B_i \{U_i(x, y) + E_i(x, y, s)\}. \quad (8)$$

Table 2. The polynomials of the error

i	$E_i(x, y, s)$
1	-
2	0
3	0
4	0
5	0
6	0
7	$-s^2 x / 4$
8	$3s^2 x / 4$
9	$3s^2 y / 4$
10	$s^2 y / 4$
11	$-s^2 xy$
12	$2s^2 xy$
13	$3s^2 x^2 + 3s^2 y^2$
14	$-2s^2 x^2 + 2s^2 y^2$
15	$s^2 x^2 / 2 + s^2 y^2 / 2$

When the wavefront under test can be expressed by Zernike polynomials, the shape analyzed by integration is expressed by equation (8).

By expanding the shape analyzed with integration in equation (8), coefficients B_i are obtained. The error $E(x, y, s)$ with integration is calculated equation (9) with coefficients and polynomials of the error.

$$E(x, y, s) = \sum B_i E_i(x, y, s). \quad (9)$$

The compensation is performed by subtracting the error $E(x, y, s)$ calculated with equation (9) from the shape analyzed with integration.

Simulation

To test the proposed method of compensation, simulation was performed. A shape used for simulation was constructed with coefficients of 21 terms of Zernike polynomials calculated

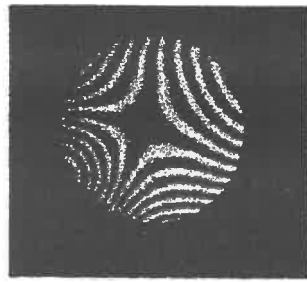


Fig. 2 Interferogram obtained with a Fizeau interferometer

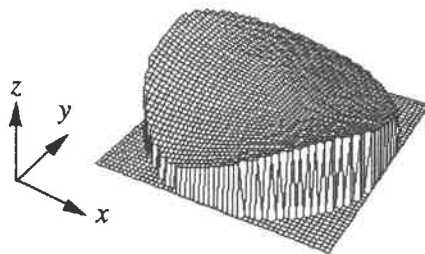


Fig. 3 A shape reconstructed with Zernike polynomials on a shape measured by Fizeau interferometer.

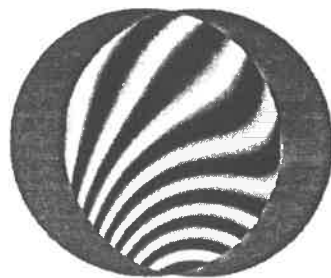


Fig. 4 Lateral shearing interferogram of the shape of Fig. 2. The shear rate is 0.2.

from a shape of a spherical concave mirror measured by a Fizeau interferometer. Interferogram obtained with the Fizeau interferometer is shown in Fig. 2. A shape constructed by Zernike polynomials is shown in Fig. 3. An LS interferogram of the shape is shown in Fig. 4. A rate of the amount of shear and a diameter of the shape was 0.2. In the analysis of the LS interferogram, a shape was reconstructed by integration. By expanding the shape in equation (7), coefficients of polynomials was found. With the coefficients, the error was calculated by equation (9). By subtracting the error from the reconstructed shape, the error was compensated. On the shape obtained by integration, PV value was 3.57λ , rms value was 0.67λ . On compensated shape, PV value was 3.64λ , rms value was 0.68λ . In the same part of the used shape, PV value was 3.64λ , rms value was 0.68λ . Compensated shape was agreed with the original shape.

Conclusion

In the analysis of LS interferogram, a method of compensation for the error with integration was presented. In simulation for the compensation, reconstructed shape with integration was expanding in polynomials of a shape to be obtained with integration. By the expansion, coefficients of the polynomials were found. And the error which calculated from coefficients and polynomials was subtracted from the reconstructed shape with integration. Compensated shape was agreed well with the original shape.

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DYNAMIC ERROR CHARACTERIZATION OF A SCANNING PROBE CMM

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INTRODUCTION

The fierce competition for market share forces manufacturers all over the world to continuously find new ways to lower production costs while maintaining or possibly improving quality. This situation leads to a never-ending quest for faster and better tools. Not only production machines but also the ones used to check the manufactured parts, the coordinate measuring machines (CMM), must reduce their measurement times to cooperate in cutting expenses. Otherwise it would be likely to spend more time checking a part than it was necessary to manufacture it.

One CMM type that has the potential for fast measurement cycles and good accuracy is the one equipped with a scanning or continuous measuring probe head. Because of this scanning capability these machines do not need to complete the cycle of contacting, then backing off and moving to the next every time a point is to be measured like the CMMs with touch trigger probes must do. Instead it simply touches the part once and scans it keeping contact following a determined path collecting a higher density of data points in a much shorter time. There are also other alternatives to fast measurements like the optical probes that are able to measure as quickly and accurately as scanning ones but cannot be used for the same applications [1].

Even though the scanning CMMs perform relatively well there is always room for improvement. For example at higher speeds there is considerable degradation of the results mainly due to dynamic problems.

A project to study, model and possibly compensate the dynamic induced errors present in a coordinate measuring machine equipped with a scanning probe was begun in 1996 at UNCC. This paper is an initial progress report on this project.

INITIAL RESULTS

A thorough literature search was carried out to verify what had been done on the subject. Nothing was found about modeling or characterization of dynamic errors of measuring machines using scanning probes. However, references from research on machines with touch trigger probes were found [3, 4]. This is understandable, since they are more widely used than scanning probes.

As a preliminary tool to investigate the dynamic errors ring gages with errors on the order of the machine's uncertainty or better were measured. The errors observed were then primarily from the

measuring procedure. Together with these tests a grid encoder was used in a similar fashion to check the behavior of the machine from a different standpoint.

For the ring gage tests two distinct diameters were used. The gages were measured on scanning mode at several speeds covering the available range. They were measured mounted at the machine table and then at a position around 300 mm higher than the table (the total Z travel of the tested machine is 400 mm). This way the machine was tested with its ram fully extended and then almost entirely retracted. By doing such experiments it was expected that the outcomes would in some way be affected by the dynamically different situations. However the results were very similar for identical conditions at the two test heights.

Additionally the ring gages were measured at table height mounted on different corners of it for several speeds. This was done to try to detect any dynamic perturbation due to the distinct position of the machine moving table but no indication was found of such disturbances.

Similar tests were executed using the grid encoder but only at table height. The grid plate was mounted on the table and the optical reader head was adapted to the probe head. The machine was commanded to move in scanning mode as if it were measuring a circle. Circular paths of different diameters were "measured" at several speeds. Although on such tests there was no mechanical contact of the probe head with a part it was expected that they could still give indications of dynamic errors. Indeed similarities could be observed comparing the results to those from tests with equivalent conditions using the ring gages.

Considering these initial findings it was decided to first address the errors of the probe head before trying to look into the structural dynamic errors. It is then necessary to mathematically model, create a software procedure and implement it in the machine control program to compensate for probe errors.

PROBE HEAD MODEL

The scanning probe has three stacked parallelograms connected by flexures that allow displacements on three directions parallel to the machine axes. This probe is a miniature measuring machine by itself with its own scales.

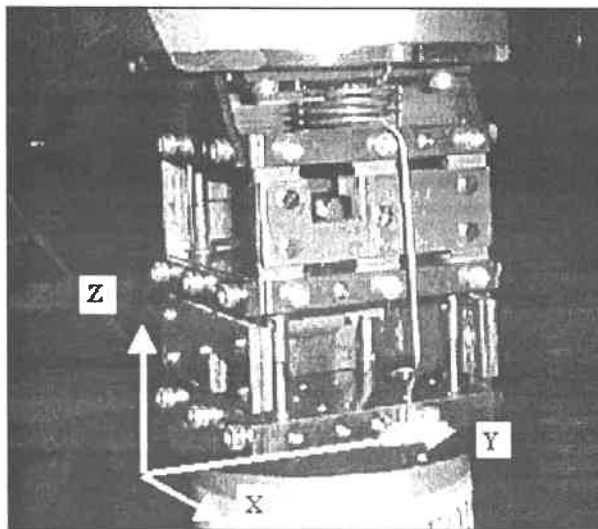


Figure 1- Probe head

The coordinates measured on a part are a combination of machine and probe scales readings.

The picture on Figure 1 shows the internal mechanism of the probe head being tested. The X

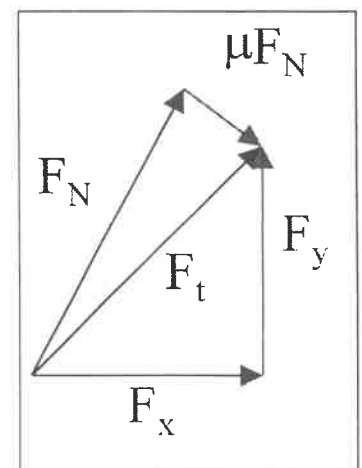


Figure 2 - Force vectors

and Y parallelograms and flexures are visible.

The physical model consists of three orthogonal stacked mass-spring-damper systems simulating the X, Y and Z deflections of the probe head flexures. To model the probe head several input variables need to be considered like acceleration, direction (type) of movement, velocity and contact forces. The three movements are initially assumed to be uncoupled.

Experiments were conducted to evaluate stiffness and natural frequency of the three probe moving axes. It was found that these values are not exactly the same, i.e., there are differences among the directions.

To measure the contacting forces acting upon the probe on an actual measurement procedure a special instrument was developed [2]. Using such device it is possible to evaluate the forces as if the machine was measuring a circular part.

Figure 2 shows the vector configuration of the forces acting on a given point during a scanning measurement of a circular path. F_x and F_y are the perpendicular force components measured by the force device, F_n is the normal force, F_t is the total force and μ is the friction coefficient. The total force is not perpendicular to the part surface because of friction.

Several experiments measuring forces, at this point only on the XY plane (parallel to the table), were already performed on the CMM using this instrument. More experiments are planned for the other machine planes using the same device. The forces were initially investigated as a function of speed and diameter. By changing diameters the idea was to detect influences of varying centripetal acceleration.

For the quasistatic deflections compensation procedure used by the machine to be effective on scanning mode the total force during measurement should be reasonably constant. However this is not

true because as speeds and/or centripetal acceleration rates increase total force values vary more and more as it was observed during the several tests conducted.

The plot on Figure 3 contains the result of such a test. The scanning speed was 50 mm s^{-1} (maximum available on the tested machine). For this test the circle was scanned twice (720°) without

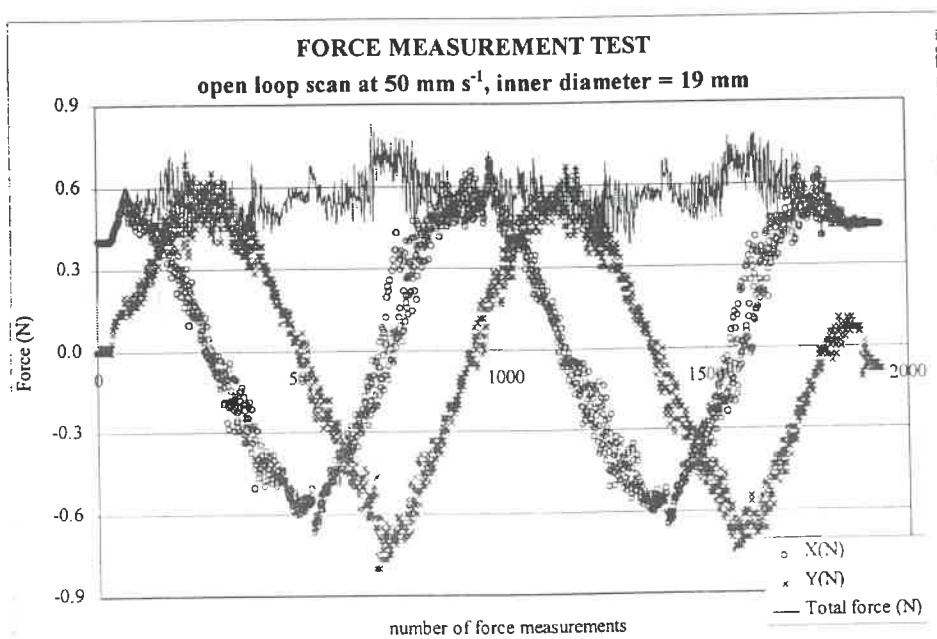


Figure 3 - Force measurement results

stopping. X and Y force directions correspond to the machine moving axes.

The variation of the total force for this measurement is almost half that of the maximum value. It can also be noted that the force patterns for the X and Y directions are somewhat different. This certainly does not facilitate a possible compensation procedure for the errors induced by these variations.

CONCLUSIONS

The variation of contacting forces noticed on the tests using the force sensor seems to be one of the major causes of the probe head dynamic errors. This can be even worse for complicated scanning paths when the probe is required to constantly change directions. Such measurements can only be done at reasonably high speed if the machine has a good dynamic behavior and enough acceleration.

The observed force irregularities are not big enough to affect the machine structure but the probe head is a much weaker component. Also the procedure of sphere calibration for probe stylus is done on point-to-point measuring mode and does not account for the varying forces that occur during scanning. So there is a clear need for some sort of compensation for the probe dynamic errors before more accurate measurements can be executed at higher scanning speeds.

Maybe these errors have never been addressed because the requirements for speedy and accurate measurements were not so stringent as nowadays. More tests are under way to provide a better understanding of the forces' behavior. Also the probe head model is under development. It is expected that this model will be generic enough to be applicable to others CMM configurations.

ACKNOWLEDGMENTS

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PERFORMANCE TESTING AND UNCERTAINTY ANALYSIS OF MULTIPLE TIP PROBING ON COORDINATE MEASURING MACHINES

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Introduction

The latest version of the U.S. standard for the performance testing of coordinate measuring machines (CMMs), ASME B89.4.1-1997 [1], contains a new test for evaluating CMM performance when using multiple stylus tip positions. The goal of this testing procedure, called the multiple-tip probing performance test, is to account for the additional errors that can arise when CMMs are used with orienting heads, probe and stylus changers, star clusters, or multiple probes. In this paper, the sensitivity of the multiple-tip probing performance test relative to errors created by multiple-tip use on CMMs is investigated. Data are presented in this paper that show that the new test lacks proper sensitivity to the errors that may occur due to multiple-tip probing use. An alternative method of analyzing the same data that is collected in the B89.4.1 multiple-tip probing performance test is presented that is more representative of the errors and more in-line with the objectives of the test. In addition, a method to use the proposed multiple-tip probing test results for estimating measurement uncertainties when using multiple-tips is discussed.

Background

According to the B89.4.1 standard, the objective of the multiple tip probing test is to understand the additional errors that may arise due to the use of multiple stylus tip positions. The standard specifically states that these errors include the uncertainty in the relative location of each of the tips due to tip calibration errors, the use of orienting heads, and probe or stylus changing systems. The error in the proper determination of the relative location of various stylus tips is also referred to as the probe offset vector error [2, 3]. The offset vector for each probe tip is usually defined relative to a common point on the CMM, such as the end of the ram axis, during the probe tip calibration process. A common way to understand the error in the offset vectors is to measure a single feature, such as the center coordinates of a fixed sphere, with each of the probe tips in question. As shown in Figure 1, if the center coordinates of a fixed sphere are measured individually with each of the three probe tips shown, the results will be three unique sphere center coordinates, i.e. (X_1, Y_1, Z_1) , (X_2, Y_2, Z_2) , and (X_3, Y_3, Z_3) . If other measurement errors are assumed negligible, then the variation in the three sets of sphere center coordinates is representative of the errors in the probe offset vectors.

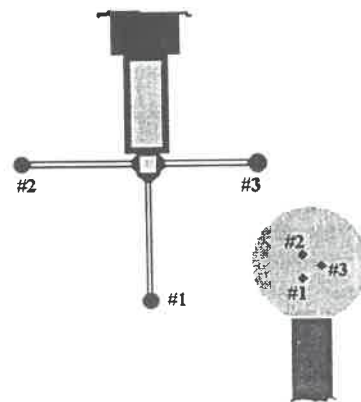


Figure 1: Variation in sphere center location due to probe offset vector errors.

In the B89.4.1 standard, the multiple-tip probing performance test prescribes that twenty-five points are to be taken on a fixed precision sphere using five points from each of five different probe tips. The five probe tips could be the result of using a fixed stylus combination (i.e. a star cluster), an articulating head,

or a probe changer. According to the standard, the twenty-five points are then used to compute a single best-fit sphere. The radius from the calculated sphere center to each individual measurement point is then computed, and the maximum minus the minimum of the twenty-five calculated radii is reported as the multiple-tip probing performance. This can also be thought of as the spherical form, or sphericity, of the twenty-five point sphere measurement.

Due to the measurement procedure used to calculate the multiple-tip probing performance, the test is sensitive to a number of errors other than just the probe offset vector errors. The results of the multiple tip probing performance tests are also a function of the machine repeatability and the point-to-point probing performance of each probe tip used in the test. In addition, the relative difference between the calibrated effective stylus ball sizes (see [3, 4]) of each of the probe tips also contributes to the resulting multiple-tip probing performance. It is the fitting of a *single* twenty-five point sphere plus the convolution of numerous errors into a single multiple-tip probing performance that results in a test that lacks proper sensitivity for the very errors that the test was designed to represent. This point will be discussed in more detail later in this paper.

Experimental Results

Using the resources of the Center for Precision Metrology at UNC Charlotte, a number of multiple-tip probing performance tests were completed on two computer-controlled CMMs under various conditions. Some of the results of the tests are shown in Table 1. The first five test results shown are from a higher accuracy CMM using a fixed stylus combination; the second five results are from a lower accuracy CMM using an articulating head.

TABLE 1: Multiple-Tip Probing Performance Test Results (all values in micrometers):

Test No.	Multiple-Tip Probing Perf.	Maximum Form	Range in Diameter	Range in X coord.	Range in Y coord.	Range in Z coord.
1	0.8	0.3	0.5	0.3	0.3	0.4
2	1.1	0.4	0.6	0.9	0.7	0.7
3	1.2	0.2	1.0	0.7	1.6	0.6
4	0.7	0.3	0.4	0.4	0.3	0.5
5	0.7	0.3	0.5	0.4	0.2	0.5
6	6.7	1.9	1.6	3.0	5.2	2.9
7	4.9	0.9	5.9	3.9	3.0	3.0
8	6.9	2.6	7.4	1.9	5.0	2.5
9	6.8	2.9	3.5	2.3	5.1	3.8
10	9.4	3.4	5.7	3.2	6.4	2.0

In addition to recording the multiple-tip probing performance, the same measurement points were also analyzed in a different manner for comparison purposes. The five measurement points that were taken on the sphere with the first stylus tip in question are used to best fit a single sphere. Repeating this procedure for the other stylus tips results in the determination of five unique and independent spheres, each one measured with a different stylus tip. From this data, the maximum to minimum range of the center coordinates of the five spheres are recorded on a per axis basis. As discussed previously, this range is representative of the probe offset vector errors. Also, the range in the measured diameters of the five spheres is noted. The maximum spherical form error (from a five point sphere) of the five spheres is additionally recorded. These results are included in Table 1.

Analysis of the data in Table 1 reveals two concerns. First, as can be seen in tests three, seven, and eight, it is possible to have measurement error in the diameter or center coordinates that is greater than the B89.4.1 multiple-tip probing performance value. In test two, for example, the range in the Y-axis center coordinate is 1.6 micrometers while the multiple-tip probing performance is only 1.2 micrometers. Secondly, as can be seen in the first three tests, for example, it is possible for the multiple-tip probing performance value to not change much even when the error in the center coordinates and diameter increase significantly. The multiple-tip probing performance values increase by 37.5% and 50% in tests two and three respectively compared to test one. As can be seen, however, the same tests show increases in the diameter and center coordinate ranges by an average of 100% to 160% with an extreme case of over 400%.

The summary of this analysis is that it is possible to (1) have errors in either the probe offset vectors or relative probe tip diameters that are greater than the resulting multiple-tip probing performance and to (2) have CMMs with significantly different magnitude of errors and have almost the same multiple-tip probing performance. In other words, the B89.4.1 multiple-tip probing performance does not satisfactorily represent the potential measurement error due to multiple-tip probing, and the test is not sensitive enough to be able to discriminate between the multiple-tip probing performance of different quality CMMs.

As was discussed previously, the reason for the problem with the sensitivity of the multiple-tip probing performance test has to do with the fitting of a single sphere with all twenty-five data points plus the convolution of numerous errors into a single test. A complete analysis of this problem is beyond the scope of this short paper. For the purposes of this paper, the problem will be summarized by stating that the best fitting process tends to average or cancel errors thereby reducing the sensitivity of the test results to some of the very error sources that the test is intended to represent, such as probe offset vector errors.

Recommendations

The issues discussed above result in this paper recommending that the B89.4.1 multiple-tip probing performance test not be used as currently stated. Instead, the recommendation is to simply use the same measurement points and compute the range of the five sphere center coordinates on a per axis basis as discussed previously. The use of the range of the center coordinates eliminates the sensitivity problems without the loss of any important information. As mentioned previously, the B89.4.1 multiple-tip probing performance test is also sensitive to other factors, such as the point-to-point probing performance and the relative stylus ball size of the individual stylus tips, but since there are other tests in the B89.4.1 standard to cover these errors, their inclusion in the multiple-tip probing performance test is redundant and unnecessary. This paper recommends focusing the multiple-tip probing performance test on representing the probe offset vector error, as that error source is not captured in any other tests in the standard. In addition, as will be discussed in the next section of this paper, the use of the range of the center coordinates as a standard test allow for straightforward estimation of the task-specific measurement uncertainty related to using multiple stylus tips.

Uncertainty Analysis

Due to the flexible and complex nature of most CMMs, estimating the measurement uncertainty for any particular feature, i.e. the task-specific measurement uncertainty, is often challenging. There has been some research done in the use of standardized test results, such as those from B89.4.1, for estimating measurement uncertainty [4,5]; however, a widely accepted and comprehensive approach to estimating

the uncertainty of CMM measurements does not exist. In this paper, the measurement uncertainty associated with the use of multiple stylus tips on CMMs is estimated using the previously described performance test that measures the range of sphere center coordinates measured with multiple stylus tips.

In the following example, it is assumed that other uncertainty contributors have been properly estimated (see [6,7]), and the focus of this paper will be solely on the additional uncertainty due to the multiple-tip use. The feature being measured is the thickness of the small block shown in Figure 2. Two alternative methods are presented, the first being the use of a single stylus tip and the second being the use of two stylus tips. For the purposes of this example, it is assumed that for the stylus length being used that there is historical performance data available that shows that the multiple-tip probing performance, in accordance to the range of centers method proposed earlier in this paper, is always less than 7 micrometers along any of the three coordinate axes.

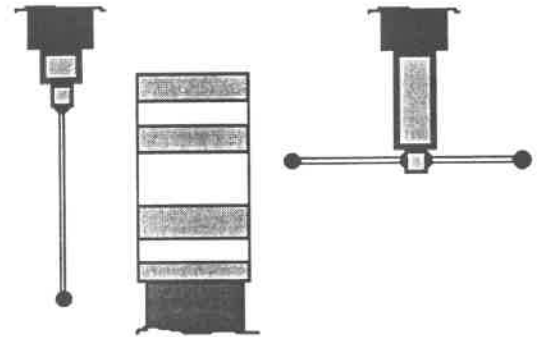


Figure 2: Two alternative stylus options for measuring the thickness of the block shown.

With this knowledge, a Type B uncertainty contributor can be estimated for the use of the multiple tips versus the single tip (assuming all other factors are the same). Since the proposed multiple-tip probing performance test result is based on the range of five centers, the use of only two tips makes it unlikely that the maximum of 7 micrometers will occur. With this information, a triangular distribution [7] is assumed with plus and minus limits of 7 micrometers. The standard uncertainty can then be computed as $u_c = 7 / 6^{1/2} = 2.86$ micrometers. This estimate can then be directly used, with a sensitivity coefficient of one, in the overall uncertainty estimate for the thickness of the block in question.

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Comparison of Simple Closure and Dual Closure Measurement Techniques on a Glass Polygon

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The question has arisen regarding the relative uncertainty of measurement between the simple closure technique and the dual closure technique for measuring the internal angles of a glass polygon. Others have shown¹ that major components of the uncertainty are the instrument to instrument variation for the autocollimators used to perform the measurement, the variation in the polygon's mirror face figure, and the between setup variation of the measurement. If high accuracy autocollimators are used to make the measurement, there is little or no advantage to using the dual closure technique over the simple closure technique. The instrument variations and setup variations dominate the uncertainty. However, if the measurement is done with lower echelon autocollimators, the dual closure technique offers more averaging of the random errors of the autocollimator readings. This can provide a better calibration than might be obtainable with the simple closure technique without resorting to the expense of a first echelon autocollimator.

In this document the results of a comparison of the simple closure technique with the dual closure technique are presented. Also, the effect of using lower echelon autocollimators is simulated by adding random factors to the existing measurement data and reanalyzing the result. The uncertainty of the simple closure method appears to be greater than the dual closure method by approximately a factor of the square root of the number of calibrated positions.

In the calibration of an optical polygon, the angles between the mirror normals of the individual faces of the polygon are the desired result. A common way to achieve this is to mount the polygon on a precision index table (called the "calibration device") and observe the angle deviation of the polygon faces with an autocollimator as each successive face of the polygon is brought into alignment with the autocollimator by rotating the index table. By appropriate design of the measurement algorithm, the deviations of the index table motion can be mathematically eliminated, leaving only the polygon deviations. Conversely, it is also possible with the same measurement setup to design the algorithm so that the deviations of the polygon are eliminated and the index table's deviations are measured instead. In a simple closure measurement technique, each segment of the polygon is compared to a single interval of an index table. A dual closure measurement compares each segment of the polygon to each corresponding segment of an index table.

One implementation of a dual closure measurement algorithm is as follows:

- ◆ With all devices set to zero, take an autocollimator reading.
- ◆ Index the calibration device to bring the next face of the polygon into view of the autocollimator, and read the autocollimator.
- ◆ Continue indexing the calibration device and recording the autocollimator readings until you have returned to the initial position of the polygon. This set of N readings, one for each face of the polygon in sequence will be called a "run."
- ◆ When a successful run has been completed, advance the calibration device one position, and realign the polygon so that its first face is within view of the autocollimator. An additional "alignment device" which may be either another index table or a less accurate rotary table, may be used to bring the first face of the polygon back into view of the autocollimator.

¹ *Advanced Angle Metrology at the National Institute for Standards and Technology*. W. T. Estler, Y. H. Queen, D. Gilsinn, And D. Pieczulewski. Proceedings of the American Society for Precision Engineering, vol 4 1991

- ◆ Take another "run" of N readings starting from the second position on the calibration table, indexing the calibration device between each reading as before.
- ◆ Continue in this manner until every face of the polygon has been compared with each corresponding calibration device position. At the end there will be a set of N^2 readings.

Arranging the data into a matrix with each "run" being a row, columns represent autocollimator readings taken on a given polygon face, and the positions corresponding to index table positions go along diagonals of the matrix.

One implementation of a simple closure measurement algorithm is as follows:

- Attach a polygon to an indexing table in the same manner as for the dual closure method.
- With all devices set at zero, take an autocollimator reading on face 1 of the polygon. (The "zero degree" face of the polygon.)
- Move the indexing table so that face 2 of the polygon is facing the autocollimator, and take a reading.
- Turn the indexing table back to zero, and realign the polygon so that face 2 of the polygon is again facing the autocollimator. Take a reading.
- Move the indexing table to bring the next face of the polygon into alignment with the autocollimator, and take a reading.
- Continue in this manner until all intervals of the polygon have been measured. In every case, the indexing table will move between the same two positions. At the end there will be a total of $2N$ readings. In effect, each interval of the polygon is being compared to a single interval of the indexing table.

The measurement technique for a dual closure measurement can be viewed as a series of interrelated simple closure measurements. Thus, with the raw data from a dual closure measurement, one can extract N subsets of these data for an N side polygon, that each constitute a simple closure measurement of the polygon referenced to a different interval of the index table.

This extraction of simple closure measurements from a dual closure measurement was performed for a 12 side polygon. The polygon was a Davidson Optronics D633-12, serial number 298. For the purpose of

this analysis three sets of simple closure data were extracted based on index table intervals 0° to 30° , 90° to 120° , and 210° to 240° . The difference between these measurements and the dual closure determination is shown here in Figure 1. The readings were all taken with a Möller Wedel Elcomat HR autocollimator. Each reading of the autocollimator is actually the average of 100 values sent by the autocollimator over a serial data line, the average of approximately 4 seconds (time) of data. The noise level of these averages in the usual lab environment is approximately

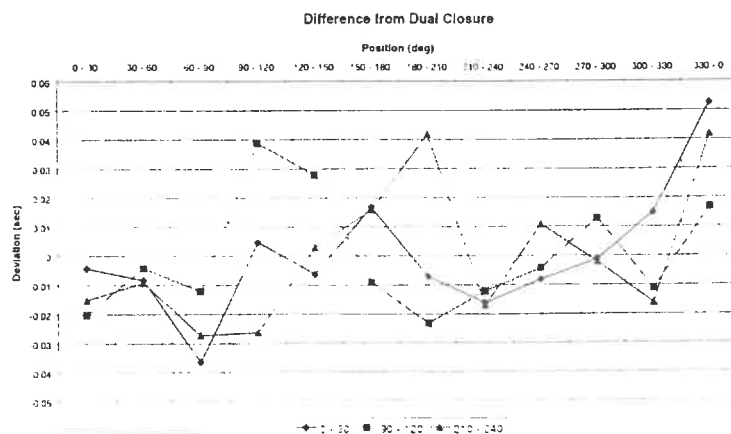


Figure 1.

0.05 seconds of arc. The maximum deviation from the dual closure measurement is about equal to the noise level of the autocollimator, approximately 0.05 seconds.

If less capable autocollimators are used to make the measurement, the amount of deviation from the dual closure calculation increases. To simulate the use of an autocollimator with a higher noise level, a randomization factor was added to the dual closure raw data, using a rectangular distribution to make these tests. This distribution was chosen simply because it would be the worst case for the simulation, and also was easy to do. A ± 0.1 , ± 0.25 , and ± 0.4 second randomization was applied to simulate autocollimators with higher noise levels than the one actually used in the test. A graph of three examples of this randomization, with ± 0.1 , ± 0.25 and 0.4 seconds of simulated noise is shown as Figure 2.

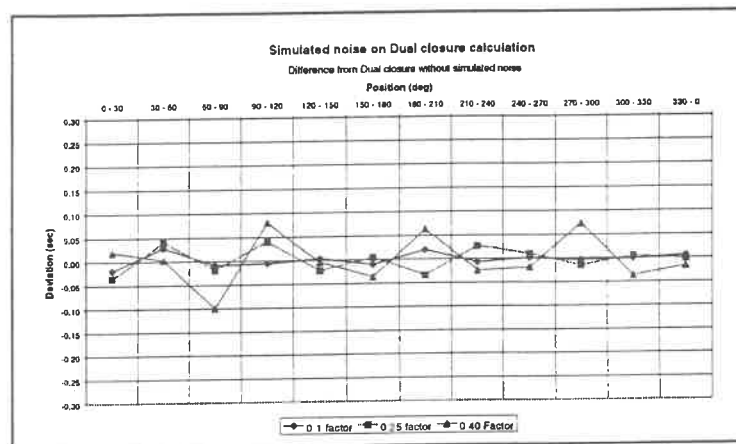


Figure 2

In this case, the dual closure measurement appears to be relatively insensitive to noise on the autocollimator readings. Table 1 below summarizes the effect of added noise in the dual closure calculations. To

Table 1.	Effect of added noise on Dual Closure calculation		
	± 0.10 noise	± 0.25 noise	± 0.40 noise
Avg RMS	0.012	0.030	0.049
Avg Max	0.023	0.061	0.096

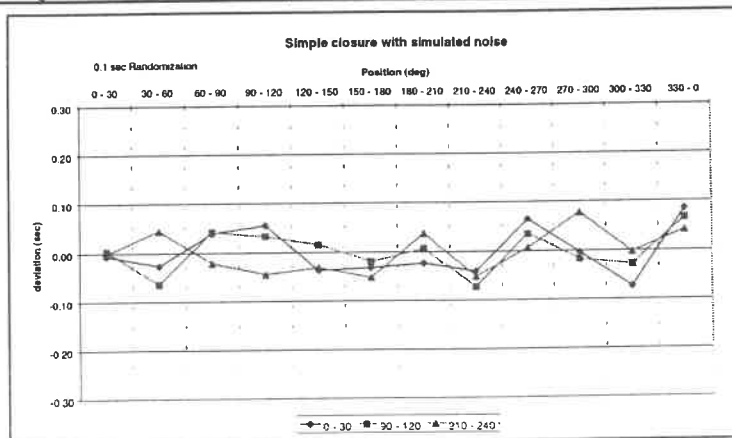


Figure 3

The effect of noise on the simple closure data is much more dramatic. Figures 3 and 4 show examples of the effect of 0.1 second and 0.25 seconds of simulated noise respectively on the three simple closure calculations taken from this data.

In this case, with ± 0.1 second of simulated noise, the largest deviation from the dual closure measurement is still fairly small, only about 0.09 seconds.

The data analyzed with ± 0.25 seconds of simulated noise, not surprisingly, shows a larger deviation from the dual closure results. In this case, the largest deviation from the dual closure measurement is about 0.27 seconds. If this exercise is repeated several times, one begins to establish a pattern to the increase in

compile this table, simulated noise of ± 0.1 sec, ± 0.25 sec, and ± 0.4 sec was added to the data ten different times each, and the average RMS deviation and average maximum deviation from the "no-noise" calculation was computed for each set.

Even in the case where ± 0.40 seconds of simulated noise was added to the data, the average difference from the raw data was less than 0.05 seconds, and the average maximum deviation in any one calculation was less than 0.1 seconds.

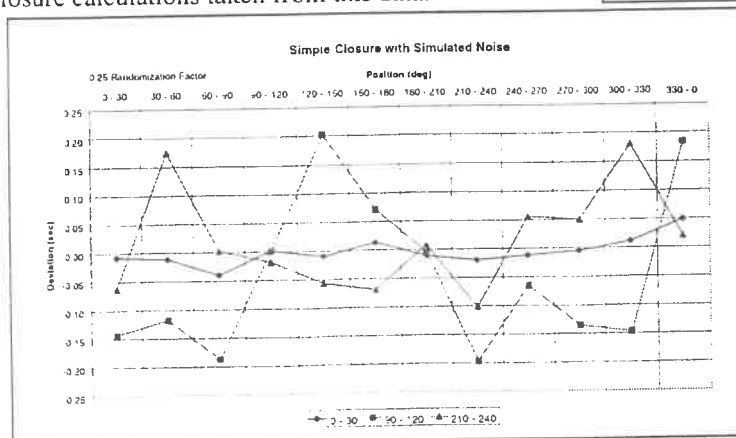


Figure 4

the uncertainty as noise is added to the data. Table 2 relates the average RMS Deviation from the dual closure result, and the average Maximum Deviation from the dual closure result for "noise factors" of 0.1, 0.25, 0.4, and 0.6 seconds. The "no noise added" column shows the average of the deviations from the dual closure result from the three simple closure calculations that were extracted from that set.

Table 2.	Effect of noise added to Simple Closure calculation				
	no noise added	±0.10 noise	±0.25 noise	±0.40 noise	±0.60 noise
Avg RMS	0.021	0.046	0.100	0.175	0.241
Avg Max	0.044	0.092	0.189	0.331	0.452

In each case, the averages are of 10 trials where the data was randomized with a rectangular distribution of the indicated width. This simulation shows a general pattern that the simple closure uncertainty is greater than the dual closure uncertainty by approximately a factor of the square root of the number of measured positions. ($\sqrt{12} \approx 3.46$ in this example). This observation is supported by theoretical work done by others.² According to reference [2], the uncertainty of the simple closure method is given by: $\sqrt{\frac{N-1}{N}} \cdot u_0$

and the uncertainty of the dual closure method is given by: $\sqrt{\frac{1}{N} - \frac{8}{9N^2}} \cdot u_0$ where u_0 = the uncertainty of the individual angle readings.

Conclusions

It appears that the increase in uncertainty of the simple closure method over the dual closure method is approximately equal to or slightly smaller than the noise level of the autocollimator used to take the readings (maximum deviation from the dual closure result). In both the simple closure and dual closure cases, the uncertainty of the measurement increases directly as the amount of noise in the readings. The simple closure uncertainty sensitivity to noise also seems to be greater than the dual closure uncertainty by a factor of the square root of the number of calibrated positions ($\sqrt{12} \approx 3.46$).

Other work³ indicates that the "best case" repeatability of the measurement of the deviation angles of a glass polygon is about ±0.15 seconds. Thus, while the deviation from the dual closure calculation is greater for the simple closure calculation with added noise, it is still not excessive until the noise level exceeds about ±0.25 seconds. This indicates there is no added benefit to using a dual closure measurement algorithm, over using a simple closure algorithm for the calibration of glass polygons. Repeatability of serrated tooth index tables tends to be better than for glass polygons, so a dual closure measurement may be justified for these items. Still, the significant increase in time required to perform a dual closure measurement over a simple closure measurement makes the relatively modest improvement in measurement uncertainty difficult to justify, if low noise autocollimators are available to make the measurement. On the other hand, the dual closure algorithm is significantly less sensitive to noise in the autocollimator readings than is the simple closure algorithm. This algorithm may be usable to get an acceptable calibration of the indexing table with a noisier autocollimator, or in a noisy environment.

Unfortunately, these tests also indicate that the uncertainty increase of the simple closure method over the dual closure method increases as the number of measured positions increases. While it may be impractical to perform a dual closure measurement on, for example, 32 positions, (at least 1024 separate readings), the simple closure method may have a larger increase in uncertainty than would be desirable. Some kind of multi-level subdivision⁴ measurement scheme may be indicated in these cases.

² *Uncertainty Analysis for Angle Calibrations Using Circle Closure*. W. T. Estler. *Journal of Research of the National Institute for Standards and Technology*, volume 103, number 2, March-April 1998

³ *Short Term Repeatability of Measurements of a Glass Polygon*. S. K. Smith. *Proceedings of the American Society for Precision Engineering*, October, 1998, (elsewhere in this volume)

⁴ *The calibration of indexing tables by subdivision*, C. P. Reeve, *NBS Internal Report* 75-750 (1975)

NIST MICROFORM CALIBRATION – HOW DOES IT BENEFIT U.S. INDUSTRY?

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Summary In microform metrology, complex 3-D surface features in the micrometer range must be quantified for their shape and size including dimensions, curves, angles, profile deviations, and alignment errors, as well as surface roughness with measurement uncertainties compatible with tolerance requirements. NIST started microform calibrations for Rockwell diamond indenters in 1994. The calibration and check standards, calibration procedures, and calibration uncertainties are introduced in this paper. NIST calibrated Rockwell indenters have been used in a U.S. national primary Rockwell hardness testing machine, and one of these is being used as a common Rockwell indenter in an international round robin comparison.

1. Introduction

Recent developments in computer science and technology, micro-electronics and precision engineering, as well as U.S. and international standardization efforts for Rockwell hardness standards, have resulted in a new area of surface metrology, called microform metrology. In microform metrology, complex 3-D surface features in the micrometer range must be quantified for their shape and size including dimensions, curves, angles, profile deviations, and alignment errors, as well as surface roughness with measurement uncertainties compatible with tolerance requirements.

One of the requirements for microform metrology comes from U.S. and international work in Rockwell hardness standardization. Rockwell C hardness (HRC) is the most widely tested material property for metal products. The United States had no national laboratory for Rockwell standards for many years. With the development of ISO 9000 quality standards, particularly for international trade, NIST was required to develop the U.S. national Rockwell hardness standard facility to support U.S. industry. One of the key points for establishing a standard Rockwell hardness scale with metrological traceability is the microform calibrations of the Rockwell indenters [1-3].

2. Technical requirements for Rockwell diamond indenters

The Rockwell diamond indenter is a diamond cone with 120° cone angle blended in a tangential manner with a spherical tip of 200 μm radius. Technical specifications include the mean tip radius, the maximum and minimum radius, profile peak and profile valley deviations, the mean cone angle, the maximum and minimum cone angle, the cone flank straightness, the holder axis alignment error, the surface roughness, and surface defects [1-8].

Two grades of Rockwell indenters are specified in ISO (International Organization for Standardization), ASTM (American Society for Testing and Materials), and CEN (European Committee for Standardization) standards [4-8]. Working grade indenters, specified in ISO 716 [4], ASTM E18-94 [6], and EN (European Standard) 10109-2 [7], are used for ordinary hardness tests. Calibration grade indenters, specified in ISO 674 [5], ASTM E18-94 [6] and EN 10109-3 [8], are

used for calibrations of the secondary Rockwell blocks. The geometric tolerance requirements for the latter are shown in Table 1. For the purpose of establishing and maintaining the national standard Rockwell hardness scale, NIST proposed standard grade Rockwell indenters [1-3] having smaller tolerances and measurement uncertainties [1-3].

3. Calibration and check standards, calibration procedures and uncertainties

A commercial stylus instrument is used at NIST for calibrating the Rockwell indenters. A manual x-y-rotary stage is used for holding and rotating the calibrated Rockwell indenter, and for crowning the stylus of the instrument on the top point of the indenter. The calibration standard for this instrument is a 22 mm radius standard ball. The check standards for the indenters include a standard wire and two ruby balls, all having 200 μm nominal radius, and a 120° angle gauge block. These check standards are calibrated at the NIST dimensional metrology laboratory with traceability to NIST dimensional standards [9].

The magnification of the instrument is calibrated by measuring the profile of the 22 mm standard ball. The calibration is certified by measuring the check standard artifacts. After this, the Rockwell indenter is calibrated at nine sections 40° apart. In each section, the stylus is first crowned on the top point of the diamond indenter. Then a 1.2 mm long trace is made across the crown of the indenter and 4800 data points are collected. By windowing on the central $\pm 100 \mu\text{m}$ part of the range and using least squares arc fitting, the least squares radius and profile deviation from the radius is determined. By windowing on the remaining left and right portions of the trace, located from -450 μm to -100 μm in the left and from +100 μm to +450 μm in the right, and using the least squares line fitting algorithms, the cone angle and cone flank straightness error are determined. The surface roughness can also be measured by using the same traced profile by selecting the appropriate filter cut-off length.

The last step is to check the instrument calibration again by re-measuring the 22 mm radius calibration ball, standard wire and angle gauge block. All these calibration data are input into our software package for calculating the calibration uncertainties for the least squares radius and angles. Based on the NIST technical Note 1297 [10], the combined measurement uncertainties are calculated as 0.4 μm for the 200 μm least squares radius measurements, and 0.01° for the 120° cone angle measurements. Both have a coverage factor of $k = 2$.

4. How NIST microform calibrations support U.S. industry

In April, 1995, a group of 11 Rockwell diamond indenters were measured by this system. These indenters showed a variation range of the least squares radii from 198.34 μm to 202.90 μm , and a variation range of the cone angles from 119.94° to 120.07° [1-3]. One of the 11 NIST Rockwell indenters, numbered 3581, was selected as the NIST master standard for the calibrations of NIST SRM (Standard Reference Materials) Rockwell C hardness blocks [11]. From 1995 to 1997, about 300 NIST SRM blocks were calibrated using this indenter, with a total of more than 3000 indentations.

In October 1997, this NIST master indenter was re-measured using the same microform measurement system at NIST. The measurement results were compared with those measured in 1995. The comparison results are shown in Table 1. The changes of this NIST master indenter

from 1995 to 1997 were within the range of the expanded measurement uncertainties. This demonstrated a long-term stability for both the microform geometry and hardness performance of the NIST master Rockwell indenter, as well as long-term measurement reproducibility for the NIST microform measurement system.

Based on the microform calibrations, a metrology approach was proposed for establishing worldwide unified Rockwell C hardness scale with metrological traceability [1-3]. A Rockwell diamond indenter calibrated at NIST and selected from the same group of 11 indenters mentioned above is being used as a common indenter in an international round robin to establish a worldwide unified Rockwell hardness scale. This "common indenter" showed very close properties, both on its microform geometry and hardness performance, to the existing used NIST master indenter [1-3]. The NIST microform calibrations support the establishment of the US Rockwell hardness scale based on metrology. The established US national Rockwell hardness scale would be compatible with the international common Rockwell hardness scale to be established using the same metrological approach.

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**Table 1: Measurement Comparison for NIST Master Standard Rockwell
Diamond Indenters Between 1995 and 1997**

Technical Requirements		Measurement Comparison Between 1995 and 1997	
Components	ISO/674 Calibration Grade	4/11/1995	10/29/1997
A. 200 μm Tip Radius - Mean Radius (μm) - Max. and Min Radius (μm) - Max. Profile Peak and Max. Profile Valley (μm)	200 $\pm$ 5 200 $\pm$ 7 + 2	199.06 $\pm$ 1.97 200.70 197.65 0.40 0.29	199.24 $\pm$ 1.19 201.58 197.41 0.45 0.33
B. 120° Cone Angle - Mean Angle - Max. and Min. Cone Angle (°) - Straightness (μm)	120 $\pm$ 0.1 120 $\pm$ 0.17 0.5 0.3	120.00 $\pm$ 0.02 120.01 119.98 0.42 0.08	120.01 $\pm$ 0.017 120.04 119.97 0.49 0.022
C. Holder Axis Alignment (°)			
D. Surface Finish Ra (μm) - Mean Ra (μm) - Max. Ra (μm)			0.0035 0.0036
CONCLUSION:		Pass ISO 674;	

A General Algorithm For Phase Shifting Interferometry Using Lissajous Figure

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Abstract

The accuracy of the phase shifting interferometry (PSI) is affected by many factors, in particular, the inaccuracies in the reference phase shift can significantly reduce the accuracy of the PSI surface reconstruction. The commonly used algorithms require that the phase shifter be shifted either at the space with some specifically selected intervals or at equally spaced intervals and deviations from these requirements will introduce big errors. Therefore the algorithms designed for these methods are very approach-oriented and have many limitations in that they usually can not process the data obtained with arbitrary phase shift steps and can not be regarded as general approaches. In this paper, a general method is proposed to process the data with arbitrary phase shift. Data from the same scan line for different fringe images are plotted to form the Lissajous figures. The phase shift steps can then be estimated from these figures by using the least squares estimations. The estimated phase shift steps are usually arbitrary and a general algorithm is then proposed to process these data. The surface information can be extracted by using the general algorithm.

Introduction

Phase shifting interferometry (PSI) has been widely used in surface metrology for almost two decades. PSI electronically records a series of interferograms while the reference phase of the interferometer is changed. The wavefront phase is encoded in the variations in the intensity pattern of the recorded interferograms and a point-by-point calculation recovers the phase. However, the existence of systematic error sources such as intensity noise, mechanical drifts, vibrations, and most importantly, inaccuracies in the reference phase shifter can significantly reduce the accuracy of PSI surface

reconstruction. The actuators used to obtain the phase shift usually contain non-linearities and hysteresis which will transform to big errors for the phase shift steps, therefore the reconstructed surface will be inaccurate. The commonly used algorithms such as three-step, four-step algorithms etc. usually require that the phase shifting steps to be multiples of $\pi/2$ and deviations from these requirements will inevitably introduce reconstruction errors. Other approaches which do not need the above mentioned restrictions require the phase shifting steps to be uniform or to be accurately known instead. Almost all existing methods have these limitations in one form or the other.

In this paper, a general algorithm is proposed to extract the measured surface information from the fringe images. Data from the same scan line of different fringe images are plotted on a plane coordinate system to form Lissajous figures. The Lissajous figures are fitted by the least squares method from which the phase shifting steps can be estimated. A general algorithm is then proposed to process the data from the fringe images using the estimated phase shifting steps. This approach does not make any assumptions and the phase shifting steps do not need to be multiples of $\pi/2$ or to be equally spaced. This will generally reduce the requirement for the control system design and the system would be more flexible.

Phase Step Estimation

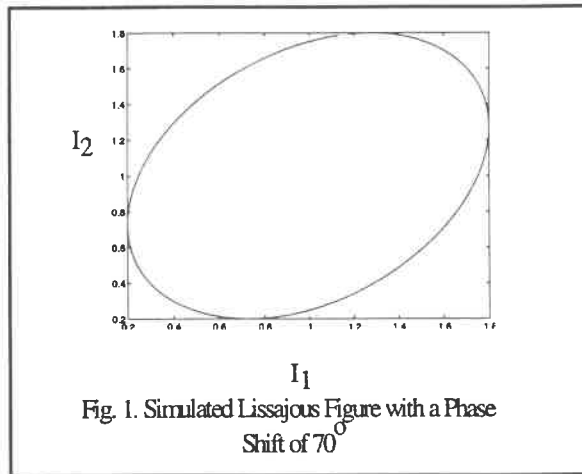
For the two beam interferometer, the fringe intensity can be expressed by:

$$I_k(x, y) = I_o(x, y) \{ 1 + \alpha(x, y) \cos[\phi + \psi_k] \} \quad (1)$$

where $I_o(x, y)$ is the mean intensity, $\alpha(x, y)$ is the fringe contrast, ϕ is the phase distribution to be measured,

$I_k(x, y)$ and ψ_k are the intensity and the relative reference phase ($k=0, 1, \dots, N-1$).

For two fringe images obtained with the phase shifting step of ψ , the intensity function from the same scan line are plotted on the X-Y plan with one set of data as X coordinates and the other sets of data as Y coordinates. These data sets can form an elliptical figure and this figure is usually referred to as the Lissajous figure (Fig.1)



Least squares algorithms are used to fit the data set to obtain the general equation of the ellipse.

$$I_1^2 + AI_1I_2 + BI_2^2 + CI_1 + DI_2 + E = 0 \quad (2)$$

The phase shifting step can be calculated from the coefficients of the equation of the ellipse by:

$$\psi = \arccos\left(\pm \frac{A}{2\sqrt{B}}\right) \quad (3)$$

One advantage of this approach is that the calculation of the phase shifting step is independent of the background light intensity. Therefore, the method is robust to the variation of the background light intensity.

Interference intensities data of a perfect flat surface are simulated and the above mentioned least squares method is used to fit the curve and the phase shifting step can be accurately estimated. Additional simulations are conducted to vary the phase shifting steps and vary the background intensity. The results also show that the phase shifting steps can be accurately estimated. It confirms the conclusion that the phase shift estimation is independent of the background intensity disturbance and this method is very robust in phase shifting step estimation.

A General Algorithm of Surface Reconstruction for Phase Shifting Interferometry

Most of the traditional algorithms for surface reconstruction are derived for some special phase shifting

steps or equal steps and can not be used in the surface reconstruction for the arbitrary phase shifting steps. What is needed is a general algorithm which can process the data for arbitrary phase shifting steps. With this kind of algorithm, it can not only lower the requirement for the control system but can also compensate for the inaccurate phase shift for the traditional phase shift interferometers.

In order to derive the general algorithm, equation (1) is rewritten as:

$$I_k(x, y) = a + b \cos[\phi + \psi_k] \quad (4)$$

$$= a + b \cos[\phi] \cos(\psi_k) - b \sin[\phi] \sin(\psi_k)$$

Manipulate it by multiplying a set of specifically selected coefficients c_k and d_k

$$\sum_{k=1}^N c_k I_k = a \sum_{k=1}^N c_k + b \cos\phi \sum_{k=1}^N c_k \cos\psi_k - b \sin\phi \sum_{k=1}^N c_k \sin\psi_k \quad (5)$$

$$\sum_{k=1}^N d_k I_k = a \sum_{k=1}^N d_k + b \cos\phi \sum_{k=1}^N d_k \cos\psi_k - b \sin\phi \sum_{k=1}^N d_k \sin\psi_k \quad (6)$$

In order to choose the values of c_k and d_k , a system of linear equations are set up:

$$\sum_{k=1}^N c_k = 0, \quad \sum_{k=1}^N c_k \cos\psi_k = 0,$$

$$\sum_{k=1}^N c_k \sin\psi_k = -1, \quad \sum_{k=1}^N d_k = 0,$$

$$\sum_{k=1}^N d_k \cos \psi_k = 1, \quad \sum_{k=1}^N d_k \sin \psi_k = 0$$

Based on this system equations, equations (5) and (6) can be simplified and the phase representing the measurement surface can be obtained by:

$$\phi = \operatorname{atan} \frac{\sum_{k=1}^N c_k I_k}{\sum_{k=1}^N d_k I_k} \quad (7)$$

The measured surface can be reconstructed by:

$$w(x, y) = \frac{\lambda}{4\pi} \phi(x, y) \quad (8)$$

Simulations

For the three-step phase shift with the steps of 0° , 90° and 180° , the calculated coefficients from the above mentioned procedure are: $c=(0.5, -1, 0.5)$, $d=(0.5, 0, -0.5)$. Put these coefficients into equation (11), we can get the commonly used three step algorithm as the special case for the general algorithm.

A flat surface is simulated and three arbitrary phase shifts 0° , 70° and 138° are simulated. The phase shifting steps are accurately estimated from fitting the Lissajous figure. The coefficients are obtained by solving the above mentioned system equations. The derived equation for extracting the surface information is:

$$w(x, y) = \frac{\lambda}{4\pi} \left(\frac{0.9060I_1 - 1.4554I_2 + 0.5493I_3}{0.2259I_1 + 0.5587I_2 - 0.7846I_3} \right) \quad (9)$$

This equation is used to reconstruct the simulated surface and the results show the perfect match of the simulated surface and the reconstructed surface.

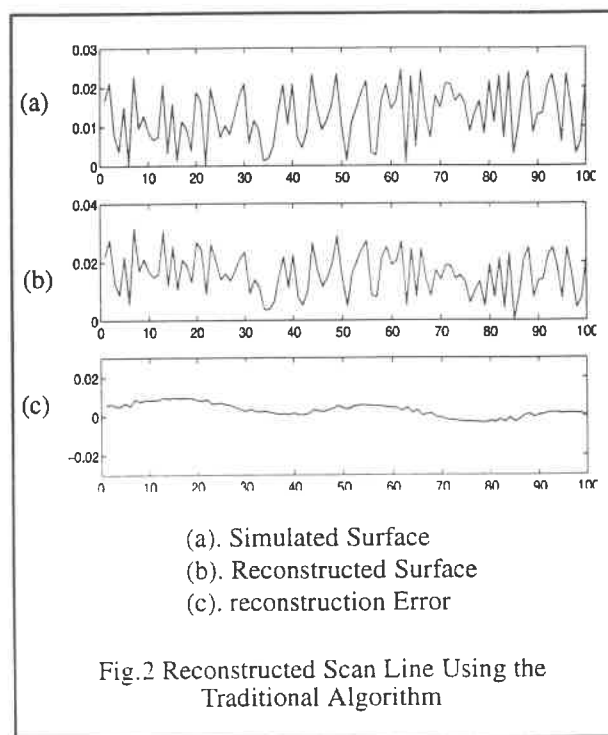
In order to illustrate the miscalibration of the phase shifter, the same simulated surface is then virtually measured by simulating three phase shifting steps of 96° , 176° , 275° which means the calibration errors are 4° , -4° and 5° respectively for the three phase shift steps. The coefficients for the general algorithm can be obtained in the same way as mentioned before. In order to avoid the

possible problem of singular matrix, the coefficient c_1 and d_2 are chosen to be 0 respectively. The derived equation for extracting the surface information is:

$$w(x, y) = \frac{\lambda}{4\pi} \left(\frac{-0.5548I_2 + 0.0980I_3 + 0.4568I_4}{0.5190I_1 - 0.4851I_3 - 0.0340I_4} \right) \quad (10)$$

The reconstructed surface using equation (10) shows perfect match with the simulated one while the reconstructed surface using the commonly used four frame algorithm shows a root-mean-square error of 2.54nm. The root-mean-square error is 1.6nm for using the new method. It shows that the new method improves the reconstruction accuracy.

A random surface is simulated and the measurement system is assumed to have a phase shifting error. Surface reconstruction using the traditional method and the new method are applied. Fig. 2 and Fig 3. show the reconstruction results. It is evident the new method generate more accurate reconstructed surface.



Experiment

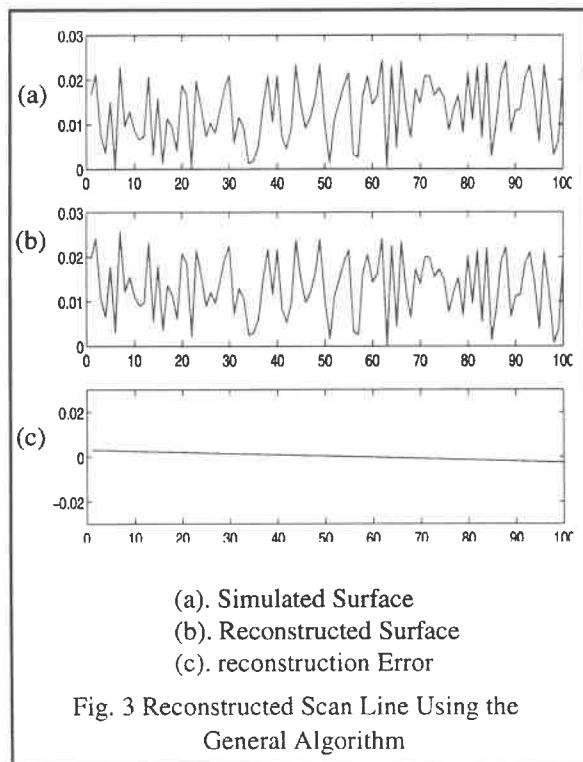
An experiment was conducted using a computer rigid disk as a specimen. Three steps were moved and four fringe images were taken. The first two steps were kept equal by providing the same increment voltage for the actuator while the last step was intentionally made smaller. The data was processed using the above mentioned algorithm. The estimated phase shifting steps are:

70.1411°, 71.1208° and 54.6999°. It is evident that even the first two steps were not equal as assumed and this was caused by the nonlinearity of the actuator and the vibration. It is not possible to use the commonly used four step algorithm to process these data.

The new method was used and the coefficients for the general equation was derived for this case:

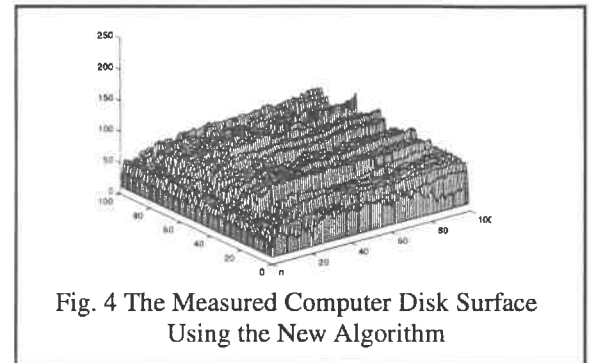
$$w(x, y) = \frac{\lambda}{4\pi} \left(\frac{0.1907I_2 - 1.3675I_3 + 1.1768I_4}{0.5246I_1 - 0.1602I_3 - 0.3645I_4} \right) \quad (11)$$

The surface reconstructed using this equation is illustrated in Fig. 4.



Conclusion

A general method is proposed to process the data with arbitrary phase shift. Data from the same scan line for different fringe images are plotted to form the Lissajous figures. The phase shift steps can then be estimated from these figures by using the least squares estimations. The estimated phase shifting steps are usually arbitrary and a general algorithm is then proposed to process these data. The surface information can be extracted by using the general algorithm. Simulations demonstrated that this general method can compensate for the mis-calibration of the phase shifter and can accurately reconstruct the surface measured with arbitrary phase shifting steps.



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The Effect of Slurry Film Thickness Variation in Chemical Mechanical Polishing (CMP)

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Abstract

In this research, it is shown that the CMP performance (material removal rate, planarization, surface roughness and defect density, etc.) is significantly determined by the slurry film thickness between the silicon wafer and the polishing pad. The slurry film thickness depends on velocity and Hersey number. Because of the dependence of material removal on the slurry film thickness and, thus, the wafer-pad contact mode, there exists a need to modify Preston's equation, a basic model used for CMP.

1. Introduction

Throughout the past decade, Chemical Mechanical Polishing (CMP) has become a leading planarization technique in the manufacture of advanced integrated circuit (IC) chips as one of the key fabrication processes. CMP is an adaptation of polishing. Even though it is widely used by IC chip makers as a reliable planarization technique, there is still a need to understand the physics of CMP from the mechanical as well as chemical point of view. In this research, the effect of slurry film thickness variation on CMP performance is studied to gain a better understanding of the mechanical aspects of CMP.

Many researchers have acknowledged the significance of slurry film thickness in CMP. The slurry viscosity, rotation speed, and wafer curvature are critical variables determining the slurry film thickness in CMP. Previous research has been conducted to calculate the variation of slurry film thickness in terms of rotation speed and slurry viscosity [1]. It was also determined that understanding the behavior of slurry film thickness in CMP will be fundamental in the investigation of material removal mechanism and the development of a process model for the CMP process [1-3]. In this study, the Hersey number [4] is considered as a main process variable to determine the slurry film thickness. Among the three parameters (velocity, viscosity, and pressure) in the Hersey number, the relative velocity is the only variable to be controlled in this study. In addition, by measuring the friction force variation between the silicon wafer and the polishing pad and referring to the Stribeck curve [5], Figure 1, lubrication characteristics in CMP can be determined. Next, the effect of slurry film thickness variation on CMP performance (such as material removal, planarization, surface defects, and surface roughness) is studied empirically. The possible dependence of the material removal mechanism on slurry thickness is also considered here.

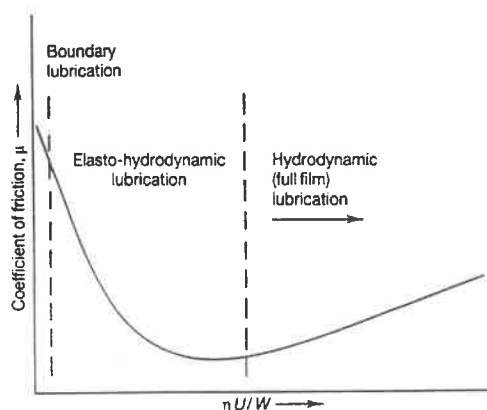


Figure 1. Stribeck curve (η :viscosity, U :velocity, W :pressure) [5]

2. Characteristics of slurry film thickness

It has been previously shown that the slurry film thickness increases with the relative velocity of the wafer, slurry viscosity, and decreasing normal pressure on the wafer. Thus, the Hersey number measured for the process, defined as,

$$\text{Hersey number} = \frac{\text{viscosity} \cdot \text{velocity}}{\text{pressure}} \quad (1)$$

is an ideal indicator of film thickness. Due to the fact that slurry film thickness influences the characteristics of lubrication, boundary lubrication, elasto-hydrodynamic lubrication, and hydrodynamic lubrication, the Hersey number is also known to be a key parameter that indicates the wafer-pad contact mode (i.e. direct contact, semi-direct contact, and hydroplane sliding).

Slurry film thickness is proportional to the square root of the wafer velocity and the Hersey number if pressure and viscosity are kept constant in the hydrodynamic lubrication regime [2][6]. The slurry film thickness for

the experimental setup used in this work was compared to that in a commercial CMP tool and the Hersey number was calculated based upon the normal process parameters, Table 1.

	The LMA machine	Cybeq 3900 [7]
Normal pressure	0.127 kPa	51 kPa
Velocity	0.2 m/s	Platen/Head = 20/15 rpm Averaged relative velocity = 0.84 m/s
Viscosity	10^{-3} N s/m ²	10^{-3} N s/m ² (assumed)
Hersey number	1.57×10^{-6}	1.65×10^{-8}

Table 1. Process parameters and Hersey numbers

Since the pressure normally utilized in a commercial CMP machine is much higher than that in our laboratory experiment, the Hersey number calculated for our process was much higher. Thus, the slurry film thickness evaluated in this study is expected to be considerably greater than the thickness common in commercial CMP tools.

3. Experimental procedure

The objectives of this experiment are to measure the friction force variation which is strictly correlated to the slurry film thickness and velocity and to identify the effect of slurry film on CMP performance parameters defined here as material removal, planarization, surface defects, and surface roughness. A schematic of the experimental setup is shown in Figure 2. The abrasive slurry, stirred by a magnetic mixer, was supplied onto the polishing pad through a peristaltic pump. Workpieces and process parameters adopted for each experiment are listed in Table 2. A load cell, with a 75 lb maximum capacity, was mounted to the plate holding the polishing pad to measure the friction force variation between the polishing pad and the workpiece. Before each test, the workpiece flatness was measured using a Taylor-Hobson profilometer to allow filtering of the geometric effect of wafer shape on the material removal rate and the friction force. The signal output from the load cell was amplified and sampled every 10 minutes using a PC with a Gagescope data acquisition system at a 100Hz sampling rate.

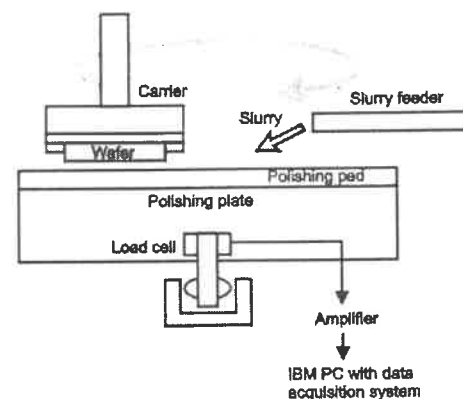


Figure 2. A schematic of experimental setup [8-9]

	Friction force measurement and material removal variation	Planarization, surface roughness, and defects
Workpiece	5 polished bare silicon wafers(<100> and p-type)	4 oxide wafers(1 μ m thick thermal oxide)
Polishing pad	IC60	
Slurry	Nalco2352	ILD1300
Dilution ratio(DI water:slurry)	10:1	
Flow rate	50ml/min	
Normal force	10-80N	80N
Velocity	3-18cm/s	3 and 18cm/s
Polishing time	10 minutes – 1 hour	10 and 20 minutes

Table 2. Process parameters for the slurry film calculation

Polished wafers were used in the friction force measurement and the material removal tests to eliminate the effect of surface roughness of the workpiece on the friction and material removal. The load cell was calibrated to measure an equivalent friction force, and the friction coefficient was calculated based on that friction force. Friction coefficients were sampled every 10 minutes for each hour and averaged to get a mean value, that is, if $\mu_1, \mu_2, \dots, \mu_6$ are the friction coefficients sampled at ten minute intervals for an hour, the mean value μ will be $(\mu_1 + \mu_2 + \dots + \mu_6)/6$.

For the material removal tests, the mass removed from the workpiece for each velocity condition was measured every 10 minutes for a hour. The Si wafer was first cleaned with an ultrasonic cleaner and dried by air jet and then weighed using a Sartorius electric balance with a resolution of 0.1 mg.

For the oxide wafers used for the tests to evaluated planarization and surface roughness effects on slurry film thickness, initial wafer curvature was measured and only oxide wafers which have the normal distance between 'peak' and 'valley' in wafer profile less than 5 μ m were selected as workpieces. After CMP, the oxide thickness

was measured with a Nanospec and the surface roughness and defects determined for each wafer using an atomic force microscope (AFM) and a Zygo laser interferometer with 0.1 angstrom resolution, respectively.

4. Results

4.1 Friction force

The friction force on the wafer surface during CMP was measured with respect to velocity in the presence of the abrasive slurry, Figure 3. In this experiment, 4 normal loads ranging from 10 to 60N were applied to the wafer during polishing. To verify the influence of abrasive slurry on the friction force, the friction force was also measured without slurry on the pad, Figure 4.

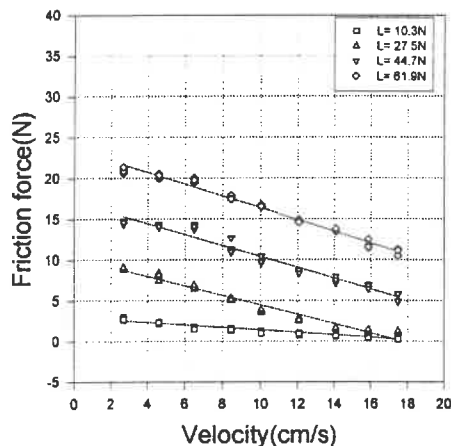


Figure 3. Friction force variation with slurry

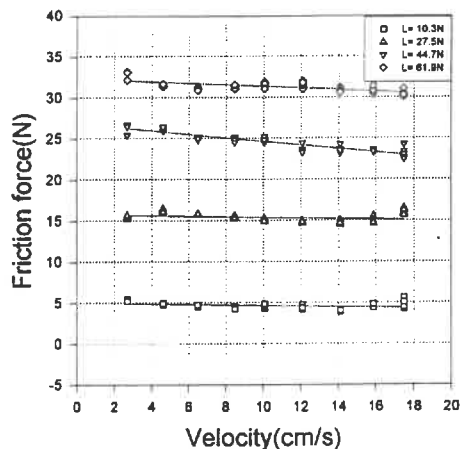


Figure 4. Friction force variation without slurry

The friction force between the wafer and the pad in dry polishing was essentially constant with velocity, Figure 4. When abrasive slurry was used, the overall absolute amount of the friction force decreased and, more interestingly, the friction force diminished with increasing velocity, i.e., increasing slurry film thickness. The variation in the friction coefficient was also plotted vs. relative velocity. The friction coefficient in dry polishing ranged from 0.45 to 0.6 and was nearly constant with respect to the relative velocity. When abrasive slurry was applied, the friction coefficient dramatically decreased to near zero as the wafer velocity increased, Figures 5 and 6.

4.2 Material removal

The material removal of the silicon wafer for a unit travel distance was measured with respect to the velocity to identify the dependency of material removal on the slurry film thickness, Figure 7. In this experiment, three normal loads were utilized.

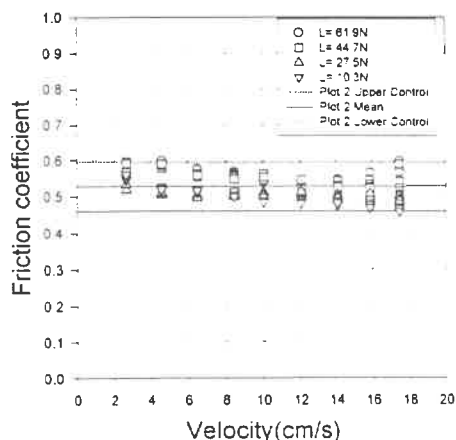


Figure 5. Friction coefficient without slurry

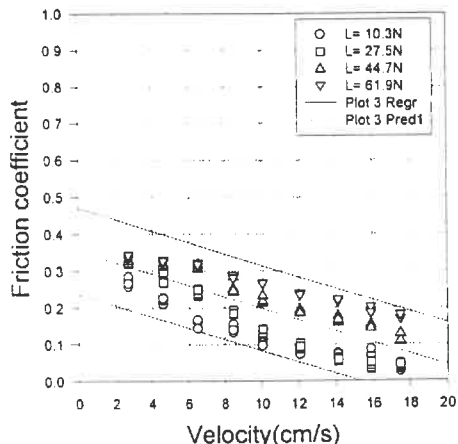


Figure 6. Friction coefficient with slurry

When polishing at the low speed, resulting in a thin slurry film between the wafer and the pad, the material removal for a unit travel distance tended to be aggressive and prominent. As the velocity increased, the material removal per sliding distance decreased. The material removal decreases as the slurry film thickness between the wafer and the pad increases.

4.3 Planarization

Twenty points of inspection along the diameter parallel to the primary flat were measured to identify the effect of the slurry film thickness on the planarization after CMP. In Figures 8 and 9, the upper and lower graphs show the flatness of the wafer before and after CMP, respectively. The oxide wafer surface polished at low speed showed better flatness and planarization than the oxide surface polished at high speed. Higher velocity created an irregular surface. To quantify the effect of the slurry film thickness on the planarization, a standard parameter, With-In Wafer Non-Uniformity (WIWNU) was calculated.

$$WIWNU(\%) = \frac{|present\ oxide\ thickness - past\ oxide\ thickness|}{mean\ of\ total\ oxide\ thicknesses} \times 100 \quad (2)$$

Equation (2) indicates that planarization is better with lower WIWNU values. From the data obtained, WIWNU calculation of the oxide surface polished by a large gap was 5% while the oxide surface polished by a small gap was 2.25 %.

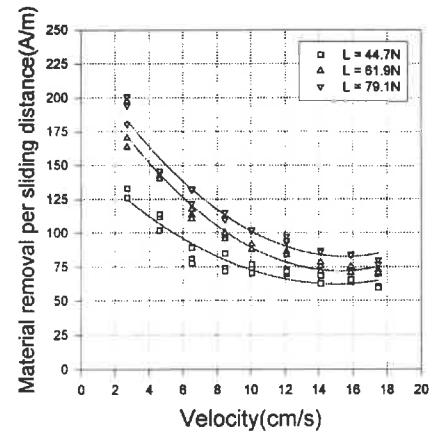


Figure 7. Material removal per sliding distance with velocity

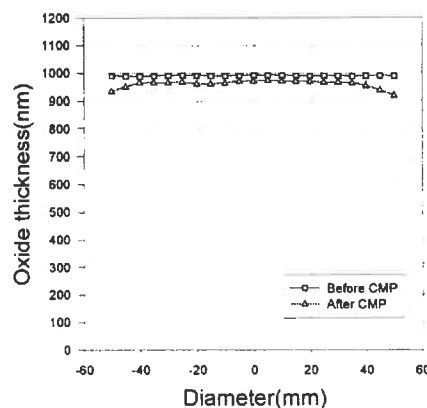


Figure 8. Planarization with a small gap

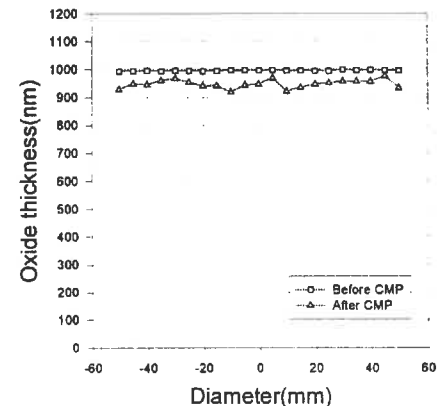


Figure 9. Planarization with a large gap

4.4 Surface roughness

Surface roughness of the oxide wafer before and after CMP was measured along with the measurement of oxide thickness variation. Three AFM images are shown in Figure 10 and surface roughness variation with slurry film thickness is in Figure 11.

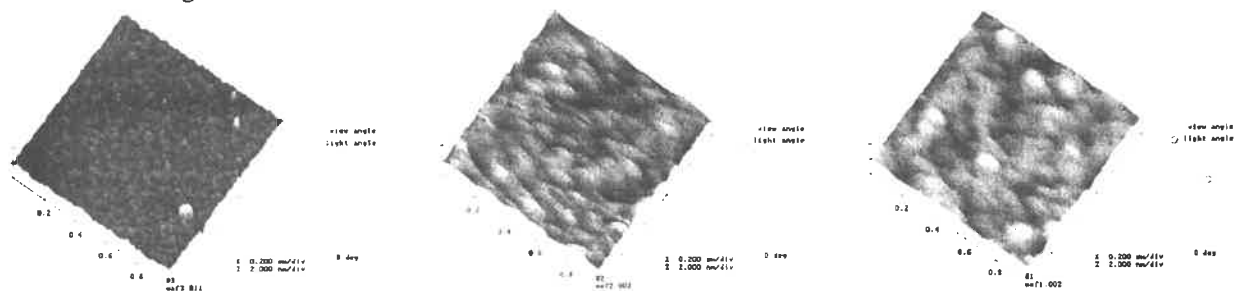


Figure 10. Surface roughness of oxide surface in 1µm x 1µm area (surface before CMP, surface polished with small gap, and surface polished with large gap)

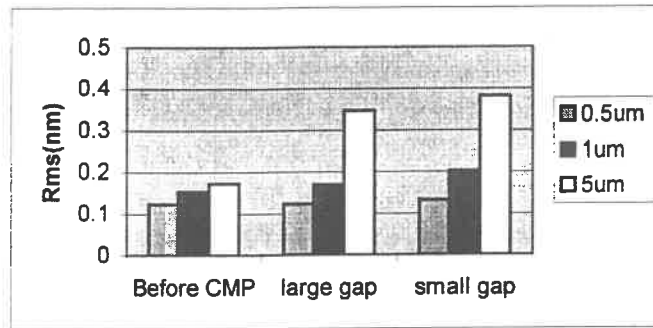


Figure 11. Surface roughness variation with slurry film thickness

From Figure 11, it can be seen that surface roughness increased as the slurry film thickness decreased and the wafer surface had more direct contact with pad. In addition to the rough surface, more surface defects such as micro-scratches on the wafer surface were detected when the wafer was polished with a small gap.

Data measured for the planarization and surface roughness study are listed in Table 3.

	Before CMP	V= 2.7cm/s (small gap)	V= 17.5cm/s (large gap)
Planarization(WIWN)	Less than 1%	2.25%	5%
Surface roughness (5μm×5μm)	Rms= 0.173nm	Rms= 0.347nm	Rms= 0.381nm
	Ra= 0.129nm	Ra= 0.271nm	Ra= 0.293nm
Surface roughness (1μm×1μm)	Rms= 0.153nm	Rms= 0.170nm	Rms= 0.202nm
	Ra= 0.114nm	Ra= 0.133nm	Ra= 0.154nm
Surface roughness (0.5μm×0.5μm)	Rms= 0.123nm	Rms= 0.123nm	Rms= 0.131nm
	Ra= 0.097nm	Ra= 0.093nm	Ra= 0.098nm

Table 3. A list of data for the planarization and surface roughness testing

5. Discussion

5.1 Lubrication analysis

The Stribeck curve, Figure 1, predicts that the friction force on the surface sliding with a thin fluid film initially decreases and reaches a minimum value as the Hersey number increases. As the Hersey number continues to increase, the friction force starts to increase again. From our experiment, the friction force between the wafer and the pad decreased as the Hersey number(as influenced by velocity) increased, Figures 5 and 7. Thus, it seems, from the Stribeck curve, that the lubrication conditions under the wafer in CMP are closer to boundary lubrication or elasto-hydrodynamic lubrication, than to the hydrodynamic lubrication. This contradicts the general view that hydrodynamic lubrication represents the behavior of the slurry film. Since the slurry film thickness in this study is believed to be considerably larger than that in an actual CMP process in semiconductor fabrication, the lubrication characteristics in a actual CMP should also be boundary lubrication or elasto-hydrodynamic lubrication.

5.2 Material removal

From this study, a possible material removal mechanism of CMP according to the slurry film thickness is proposed here, Figures 12 and 13. Although it is believed that the wafer-pad contact mode generated from our experiment is either direct or semi-direct contact mode, a possible material removal mechanism according to the wafer-pad contact mode is proposed, including the hydroplane sliding mode.

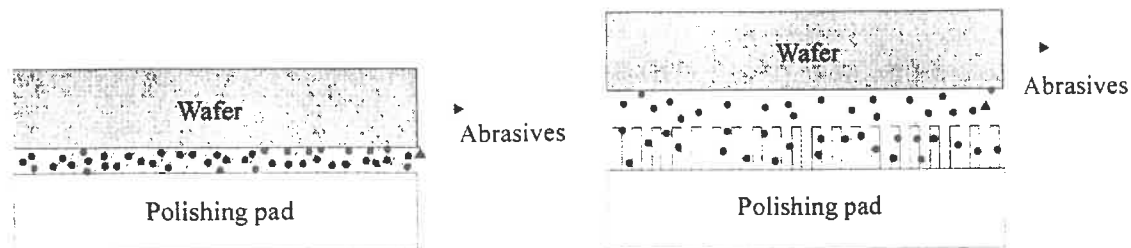


Figure 12 A proposed schematic of material removal in CMP for the slurry film thickness

If the slurry film thickness is small and the contact mode between wafer and pad is either direct or a semi-direct, the material removal is aggressive and mechanical removal (nano-plowing and nano-scratching) is dominant

compared to the chemical removal. Due to this dominant mechanical removal, there exists a greater possibility of generating micro-scratches on the wafer surface during CMP. A rough surface is generated compared to a surface finished by a chemically dominant action. However, due to the semi-solid contact with the pad, global planarization of wafer is superior.

As the slurry film thickness increases and the contact mode between wafer and pad varies from direct or semi-direct contact mode to a hydroplane sliding mode, the material removal per sliding distance decreases. This indicates that chemical removal by the slurry plus minor mechanical removal becomes the major material removal mechanism. Due to the dominant chemical removal and reduced mechanical removal, the surface roughness is improved and there is less possibility for surface defects like micro-scratches, for example. It is more difficult to achieve global planarization in this mode since the chemical removal is the primary material removal and is more isotropic in directionality, as in an etching process.

The evidence showing the dependence of material removal on the wafer-pad contact mode suggests the need to modify Preston's equation. Furthermore, it may prove useful to optimize the slurry film thickness to balance the mechanical with chemical removal to maximize the material removal and optimize planarization(as measured by WIWNU) in CMP.

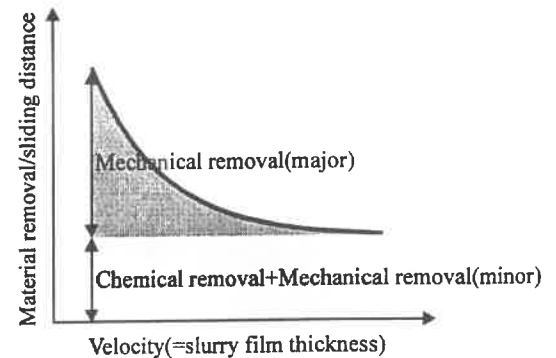


Figure 13 Mechanical and chemical removal for slurry film thickness

6. Conclusions

The following conclusions can be drawn from this work:

- The slurry film thickness between the wafer and the pad is proportional to the velocity of the wafer and the Hersey number.
- The effect of slurry film thickness variation on the CMP process performance (material removal, planarization, surface defects and roughness) is significant.
- There is a need to modify Preston's equation based upon the dependence of material removal on the slurry film thickness
- It may be useful to control and optimize the slurry film thickness to maintain process stability

7. Acknowledgements

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Precision optics fabrication using magnetorheological finishing

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In magnetorheological finishing (MRF), a workpiece is installed at some fixed distance from a moving wall, so that the workpiece surface and the wall form a converging gap. An electromagnet, placed below the moving wall, generates a non-uniform magnetic field gradient in the vicinity of the gap, normal to the wall. Magnetorheological (MR) polishing fluid is delivered to the wall just above the electromagnet pole pieces. The MR fluid is pressed against the wall by the magnetic field gradient, acquires the wall velocity, and becomes a plastic Bingham medium before it enters the gap. Thereafter, a shear flow of plastic MR suspension occurs through the gap, resulting in the development of high stresses in the interface zone and thus, material removal over a portion of the workpiece surface.

For precision finishing, MRF employs a computer program to determine a schedule for varying the position of a rotating workpiece through the polishing zone. Because the tool is a fluid that works on only a portion of the workpiece at any given time, this polishing tool may be used to finish complex surface shapes like aspheres that have constantly changing local curvature. For a spherical surface, a rotating workpiece is swept about a pivot point that is located at its center of curvature, and dwell time is varied at each radial and axial position to enable deterministic material removal from the workpiece surface.

An advantage of MRF is that the polishing tool does not wear, since the recirculated fluid is continuously monitored and maintained. The technique requires no dedicated tooling or special setup. Integral components of the MRF process are the software, the 4-axis PC based CNC platform that executes motion and programmable logic control, the MR fluid delivery and recirculating/conditioning system, and the magnetic unit with integrated fluid carrier surface. A computer algorithm generates machine control programs for MRF. Inputs to this software are the interferometrically derived initial surface error, and the material removal function obtained on a test part that is identical to the part to be polished. Outputs are the machine program (instructions in the form of a velocity schedule) and a prediction of the final surface figure.

The requirement for the machine platform is to execute a custom program that is designed to sweep a plano, convex or concave lens through a ribbon of MR suspension. It accomplishes this task by using four active axes, X-, Z-, B-, and A, which move simultaneously in synchronization and are responsible for sweeping a rotating lens through the MR suspension about a virtual pivot point. The purpose of the delivery system is to supply the polishing zone with recirculated fresh MR suspension. The delivery system also comprises an in-line conditioner which continuously senses and stabilizes the suspension through a closed loop feedback circuit. Very high precision optical surfaces, with figure accuracy exceeding 30nm p-v and surface roughness under 10 Angstroms rms, may be produced using commercially available magnetorheological finishing systems.

Keywords:
magnetorheological finishing, MRF, precision polishing, aspheres, deterministic processing

"DUCTILE MODE" MACHINING OF FERROELECTRIC MATERIALS

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Introduction

Ferroelectric ceramics and single crystals have attained considerable importance in a widening range of applications. Their exceptional combination of properties include strong piezoelectric, pyroelectric and electro-optic effects. One of the most important and widely-used groups is the lead zirconate titanate solid solution series (PZT). PbTiO_3 and PbZrO_3 are perovskites which form PZT solid solution ($\text{PbZr}_x\text{Ti}_{1-x}\text{O}_3$), over the whole composition range [1]. For $x = 0.52$, the piezoelectric, elastic and dielectric constants rise to their maximum values. This boundary is termed the morphotropic phase boundary (MPB). A wide range of products relies on the high-precision fabrication of piece-parts using ferroelectric ceramics, including: ultrasonic medical imaging transducers, ink jet printing heads, pyroelectric IR detection arrays, and optical modulation systems. Not only is precision important, but also the degree of sub-surface damage needs to be minimised because this is known to compromise performance by engendering depoling. It also leads to in-service problems due to ageing and noise effects caused by the movement of damage-induced domain walls and cracks. Little information is available on the mechanical behaviour of the group of ceramics which have compositions near the MPB [2]. The contemporary technology that has been used to machine these materials has been diamond sawing and grinding followed by free-abrasive lapping and polishing. The objective of this study was to investigate a 'ductile mode' machining process for this group of ceramic materials reducing sub-surface damage and thus eliminating the need for lapping and polishing, leading to reduced costs of production.

Material Preparation

4mm thick disk shaped samples were diamond sawn from cylinders of PC4D (hard) and PC5K (soft) MPB PZT materials produced by Morgan Matroc Ltd (Unilator Division). These were lapped and polished on a Logitech PM4 polishing unit fitted with a PP6 vacuum jig. The final flatness was to within better than half a wavelength of HeNe laser light ($\sim 300\text{nm}$). Surface preparation was a two stage procedure. Initially, samples were lapped flat with $3\text{ }\mu\text{m}$ alumina in water slurry on a cast iron radially grooved lap. The second finishing stage consisted of polishing for one hour with colloidal silica on a polyurethane pad. The polishing pressure was 100g/cm^2 ($\sim 10,000\text{ N/m}^2$) throughout. Samples of three different conditions of each composition were prepared. The conditions were a) poled in-plane, (or parallel to the surface), b) poled perpendicular to the surface and, c) unpoled. By observing the $\{002\}/\{200\}$ X-ray diffraction (XRD) reflection doublet it could be seen that very little domain re-orientation had taken place in the near surface due to the final polishing.

Quasi-static Indentation and Ruling

The crack system obtained in quasi-static indentation can serve as a basis for descriptions of material behaviour in processes such as machining. The response of materials to indentation provides information about wear, machining damage, fracture and yield strength. However, machining damage differs morphologically from isolated indentation damage with regard to the configuration of the plastic deformation zone and associated cracks, as well as the multiplicity of neighbouring damage sites [3]. The quasi-static indentation technique was used in this study to investigate the behaviour of PC4D and PC5K ceramics. Vickers indentation experiments were performed on a Matsuzawa Seiki MHT-1 micro-hardness tester with loads ranging from 10 to 500 grams. Six indentations at each load were made and analysed to obtain the hardness and fracture toughness values for both compositions. The values of PC4D hardness and fracture toughness were consistently higher than for PC5K. Poling direction had an

important effect on fracture toughness results. Threshold loads and cracks were calculated and the results were found to agree with the Hagan [4] model for brittle materials. Yield Stresses were calculated based in the model proposed by Studman, Moore and Jones [5] and the results considering the stresses parallel (E_{33}) and perpendicular (E_{11}) to the polar axis.

Poling direction relative to the sample surface	Stress direction relative to polar axis	Yield Stress PC4D (GPa)	Yield Stress PC5K (GPa)
Unpoled	N/A	1.46	1.12
Poled Parallel	E_{33}	1.46	1.16
Poled Perpendicular	E_{11}	1.57	1.32
Poled Parallel	E_{11}	1.62	1.39

Table 1 Yield Stress for PC4D and PC5K Ceramics

Complementary to quasi-static indentation the process of single point diamond ruling as for diffraction-gratings was used for this study [6]. Tribological studies of ruling have demonstrated its relationship to quasi-static indentation [6] and it has been shown that ruling is one of four fundamental material ablation processes categorised according to pre-set loading or depth and singularity or multiplicity of tool-points [7]. Seven different loads from 21.5 to 113.4 grams (0.2109 to 1.112 N) were used in the ruling machine which used a hatchet-shaped tool. All samples were machined with groove direction parallel to the surface. For the samples poled in-plane, or parallel to the surface, two machining conditions were analysed: a) parallel to the poling direction and, b) orthogonal (90 degrees) to the poled direction. Three maximum ruling speeds of 22, 39.8 and 63.7 mm/sec were used for each load. The movement of the leadscrew was calibrated to maintain an index movement to avoid interference between adjacent grooves. XRD analysis using radiation $\text{CoK}\alpha$ was undertaken before and after ruling and the degrees of domain reorientation were compared. The ductile/brittle threshold depth of cut dependent upon material mechanical properties and the orientation of poling. PC4D exhibit a more ductile response when comparing with PC5K exhibiting this response over a range of loads and speeds. Using the model proposed by Bifano [8] was possible to predict the critical depths of cut for ductile grinding as shown in Table 2.

Poling direction relative to the sample surface	Stress direction relative to polar axis	Critical Depth of Cut of PC4D (μm)	Critical Depth of Cut of PC5K (μm)
Unpoled	N/A	0.3	0.4
Poled Parallel	E_{33}	0.6	0.7
Poled Perpendicular	E_{11}	0.4	0.3
Poled Parallel	E_{11}	0.2	0.1

Table 2. Critical Depth of Cut after Bifano (1988)

These results show good agreement with the figures actually observed for PC4D and PC5K poled parallel to the surface and provide a basis for other machining processes to be interpreted.

Diamond Turning

For some years single point diamond machining has been used routinely in many applications. Under normal engineering conditions diamond-turning of brittle materials is considered to be a crushing process producing surfaces characterised by conchoidal fractures and deep damaged layer [9]. Ductile machining phenomena have been observed in brittle materials in which a light cutting force has been applied. In such observations instead of brittle damage, material was seen to pile up either side of the machined groove [10]. This phenomenon was described for single point diamond turning by Puttick et al.[11], and it is apparent that, for cuts below some critical value (critical depth of cut), together with appropriate machining conditions, glass and other brittle materials can be machined in a ductile manner. Some experimental methods have been used to determine the critical ductile-brittle transition in single point diamond turning experiments by machining a wedge grooved surface[12], and

more recently by the shoulder analysis technique[13]. The latter method requires removing the tool from the surface during a cut, leaving a shoulder in the shape of the tool radius. The work established general trends of the ductile-brittle transition as function of tool geometry and machining parameters (i.e. cutting speed). Sintered PZT ceramics have been single point diamond machined by Nah [14] and it was observed that the material composition was the most influential aspect in that tests. Feed rates and depths of cut were found to be the most important machining parameters when single point diamond machining these materials. A Taguchi design experiment using two variables with three levels was prepared. The method was used in order to evaluate the sensitivity of the most important machining parameters based on previous tests[14]. Two variables, feed rates and depths of cut, with three levels were chosen based on those used by Nah, although those materials had a higher level of porosity and therefore a lower mechanical strength. Using the shoulders produced in the unpoled ceramics and the equation proposed by Blake [13] the critical depths of cut calculated are shown in Table 3.

	PC5K	PC4D
Critical depth of cut (μm)	0.043	0.170

Table 3 Critical depth of cut using shoulder technique after Blake (1988)

Some systematic changes of the depth of cut were unavoidable although a tool positioning system controllable to tens of nanometres was used. These changes were calculated from the known sample form and used to investigate different conditions of cut per revolution. The results of the depth of cut show good agreement with the results previously calculated. The diamond tool however show a high level of wear during the tests thus indicating a reduced possibility to use the turning process as a production tool to machine PZT ceramics.

Grinding

The surfaces of precision ceramic components used in engineering applications are often finished by grinding. Fine grinding processes are considered to generate higher compressive residual stresses in a machined ceramic surface which is beneficial in increasing the flexural strength and resistance to in-service strength deterioration by mechanical contact of the workpiece materials. The efficient grinding of high performance ceramics requires careful selection of the operating parameters to maximise material removal rate whilst ensuring surface integrity. The careful selection of these parameters can reduce the overall cost of production which can be extremely high when machining ceramic materials [15]. A basic Taguchi array with its respective runs was used for the PZT ceramic tests. Samples were prepared unpoled for both PZT compositions and two diamond resin bonded wheels were used for initial tests. The grinding machine had a means of truing the diamond wheel to a sub-micron level and also to truncate the grits to a uniform height to permit ductile regime grinding. The abrasive grit size and depth of cut were identified as the most influential parameters in the initial results. The glazing of the wheel during the tests was also identified as an important factor. Using the results of the previous machining tests and the initial grinding tests, a set of new tests using the ELID process and a metal bonded wheel was designed. The results of this final tests show that the R_a surface average measured with a Wyko White Light profilometer for all samples was 5.27nm better than all average results obtained from the polishing tests using colloidal silica and all the samples ground using the Electrolytic In-process Dressing (ELID) process had a very shiny surface finish [16]. The depth of the damage measured using X-ray Diffraction (XRD) analysis was about $2\mu\text{m}$ which is a value much better than the $11\mu\text{m}$ obtained during turning tests and a similar value observed by Shore [17] in glass materials.

Concluding Remarks

Quasi-static indentation techniques associated with ruling studies have been shown to provide some important insights into the mechanisms arising during the machining of PZT. Ductile behaviour was observed under some conditions, and poling procedures were found to change the machined behaviour of the same material composition. XRD analysis showed that the domain switching is associated with the 'ductile' machinability with this group of ceramics. Ductile mode machining was achieved in both soft and hard PZT compositions when single point diamond turned. The hard material compositions showed a more stable behaviour than the soft material and they were easier to machine in a ductile mode. XRD tests indicate that negative rake angles had a beneficial effect on

the textural effects, reducing domain switching. Increases in feed rates and depths of cut reduced the domain switching due to the generation of brittle damage. Although the cutting forces were very small, the cutting areas were also small which increased the pressures on the tools over the surface. Diamond wear rate is also high when machining these materials, which together with the high levels of switching domain suggest that the single point diamond turning process should be used only for special applications when machining PZT ceramics. In order to better distribute the force applied and overcome the problem of the wear rate of the diamond tool, a multi-point diamond machining process e.g grinding is required. The ductile grinding process using a fine grit diamond metal bonded wheel and the ELID process proved to be a satisfactory method for producing PZT devices with a low level of sub-surface damage. Using this method it is possible to avoid undue loading of the wheel which seriously damaged the surface finish in previous tests. In the grinding process the swarf of PC4D and PC5K materials adhere very strongly to the diamond grits thus building up an edge on the grits in the wheel. This observation corresponds to that made in the ruling and turning tests with PZT ceramics. It was also observed that there was a large reduction in the normal forces when the finer grit wheel and a smaller depth of cut were used. There was also a clear relationship between the abrasive grit size and the crystallographic texture in the near-surface region. The reduction of grit size together with an increase in the density of the grits on the wheel brought the forces to a very low level thus contributing to the achievement of ductile mode grinding. Regions of pure plastic behaviour were therefore observed about the grinding grooves as also observed in the ruling and turning studies.

Acknowledgments

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Accuracy Optimization of Machine Tools under Acceleration Loads for the Demands of High-Speed-Machining

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Abstract:

The increasing influence of tool path deviations based on structural deflections arising from acceleration loads requires modified investigations of machine tool accuracy. A method has been developed to investigate the path accuracies of machine tools in time and frequency domains. Results of applying the approach to parallel machine tool concepts are also shown.

In contrast to investigations using Finite Element Methods (FEM) [1] focused on structural components only, the proposed method evaluates the influence of the drives in combination with controllers and structure on tool trajectories with moderate computational effort. This approach is also useful for investigating the dynamic behavior of parallel machine tool structures.

Keywords:

Simulation, Optimization, Path-Deviation

1 Introduction

The application of High Speed Cutting (HSC) technologies necessitates high axis accelerations due to process conditions. These high accelerations force the process to fall under the realm of High Speed Machining (HSM) working conditions. By this, forces due to inertial loads are significantly greater than forces due to process loads. HSM requires feed axis drives and machine structures capable of driving the tool at adequate feed rates along the milling path. Today's machine tools are complex systems consisting of structures, drives and controls. Therefore optimization of these systems can be quite time demanding.

A pre-optimization of the machine tool has to take place in the design phase based on models that include the significant influences and which are suitable for advanced dynamic accuracy monitoring. In order to minimize the computational effort that is required to perform these numerical investigations, the applied

models have to be sufficiently simplified but still have the capability of showing the relevant effects. Such simple yet meaningful models lead to a reduced development duration through estimation of the future system properties.

This paper describes the components and the application of a simple but useful approach to evaluate the static and dynamic properties of HSM machine tool systems in a condensed form.

2 Rigid-body approach

Results of experimental modal analysis show the dominant effect of guideway-stiffness on structural deformations in the range of 5 – 300 Hz. Therefore, the machine tool modeled is sufficiently represented by rigid bodies coupled by so-called Stiffness-Damping-Elements (SDE), including the guideway properties. Stiffness and damping of SDEs can be estimated experimentally of the physical properties of relevant machine components. This structural model is then completed by additional components representing drives and controls that are applied to perform the actuation of the machine tool.

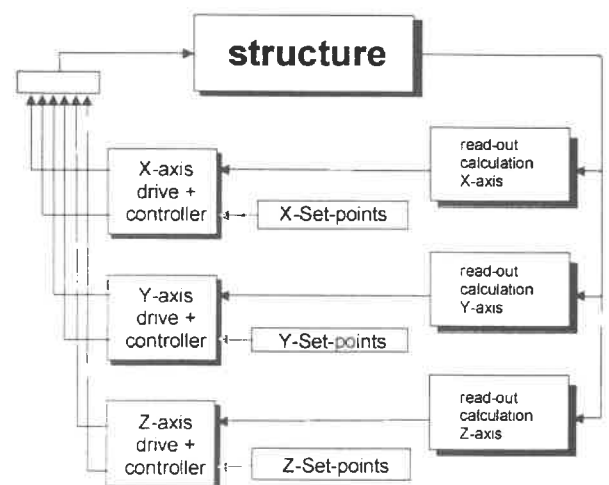


Figure 1: Matlab/Simulink structure of the machine tool model.

The following components are used to form the complete model of the machine tool using Matlab/Simulink [2]:

Structure

Arrangement of machine tool sub-assemblies, their masses, moments of inertia and the attachment points of the SDEs which represent the coupling with the other elements are required to define the system. The mass, stiffness and damping matrices are derived from the physical parameters of the structure, and then cast into the state-space-representation for use in the Matlab-Simulink-model.

Measurement systems

The measurement-system (linear-scale-) read-outs are calculated using the relative positions of reading-head and scale in space. These read-outs are then fed into the control blocks.

Drives and control

Characteristic properties of the drives and controls are frequently used for simulative investigations of feed-drives [3]. The drives act as external forces on the rigid body system.

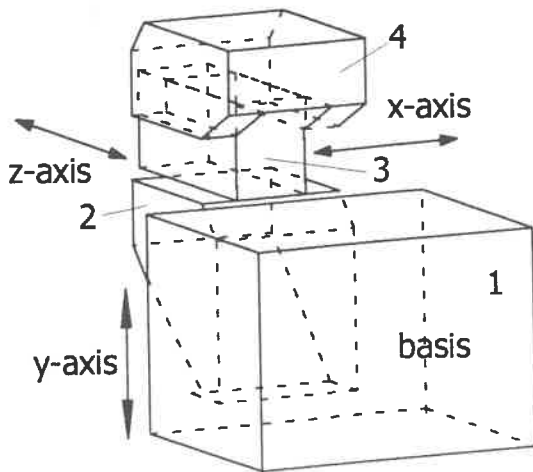


Figure 2: 3-Axis machine tool configuration, used in the examples.

The generation of these models can be performed in an efficient manner utilizing the basic geometric, physical and technological data already available at an early stage of machine tool product development.

3 Application in Time-Domain

In the time domain, positioning behavior as well as contouring performance can be analyzed according to the six degrees of freedom that characterize the relative motion of two bodies (in this case the tool and workpiece).

The simulation output in time domain offers the simulation of results which normally only can be obtained with the Heidenhain grid-encoder measurement system which has been used for a

large number of measurements in the past and which led to a better insight into the path deviations occurring at high accelerations[4]

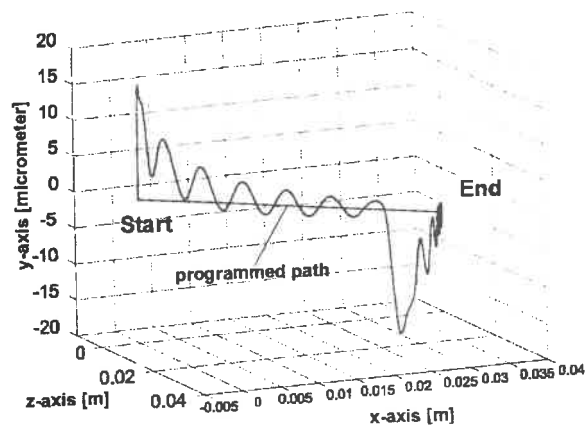


Figure 3: Simulation result: Deviations in y-direction on a linear motion in x-z-plane.

Figure 3 shows the lateral path deviations when contouring along a straight line. Strong path deviations can be easily observed especially during the acceleration and deceleration phases [4]. Investigations like this help to minimize the influence of machine tool structure dynamics on path deviations such as axis cross-talk.

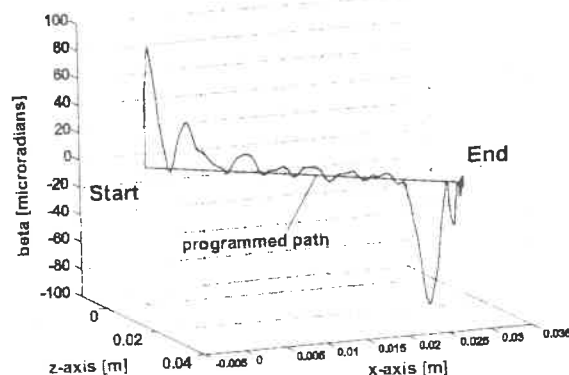
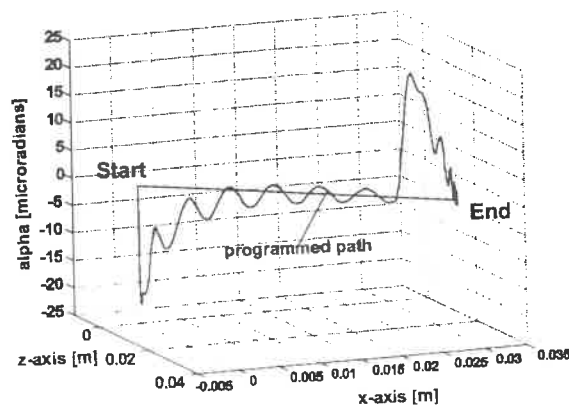


Figure 4: Simulation result: Rotative relative motion around x-, y-axis respectively on a linear motion in x-z-plane.

When focusing on the axis performance in only the direction of main movement, the effects across the path, which can later be observed as part of the surface geometry, remain hidden. Even under machining tests the structure-based deviations may not be detected due to process influences that are induced by actual process parameters. As can be seen in figure 4, the tool (fixed to body 4, figure) performs a rotation relative to the table (body 1) due to acceleration loads placed on it. Comparison of these values with given limits allows the evaluation of system dynamic capabilities, which are given by structural boundary conditions.

Measurement read-out vs. TCP-displacement

The discrepancy between the positioning behavior that is observed when using the measurement system read-out and the actual TCP (tool center point) relative motion is shown in figure 5.

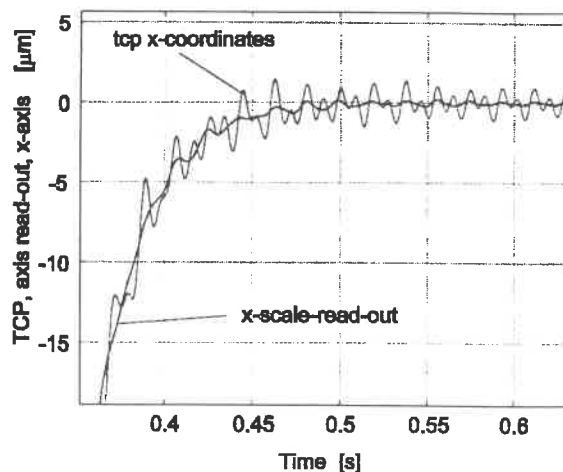


Figure 5: Simulation result: Measurement-system read-out vs. TCP-Positioning.

The positions strongly differ due to large offsets between the centers of inertia, the drive attachment points, the locations of the linear scales, and the tool-workpiece contact area. The estimation of these effects at the design stage helps to avoid the need for time-consuming re-work that is sometimes required after the machine has already been built.

4 Application in Frequency-Domain

Data analysis in the frequency domain is used to identify eigenfrequencies, mode-shapes, and dynamic and static stiffnesses. It must be noted that this approach focuses mainly on the TCP-behavior and not on the investigation of single structural components.

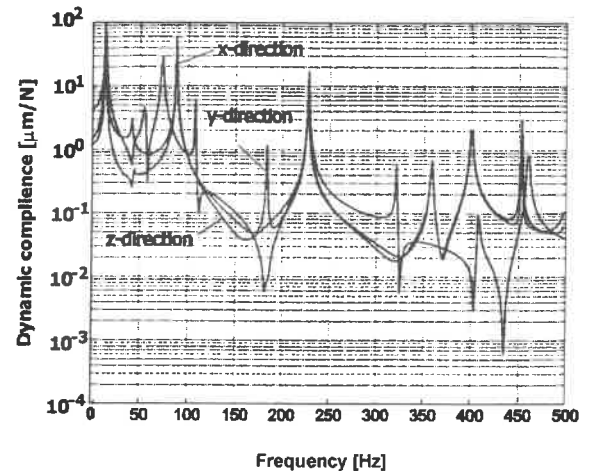


Figure 6: Dynamic stiffness calculated at TCP, in x-y- and z-direction

Figure 6 shows the dynamic stiffnesses given by the structural loop calculated at the TCP in different directions. The modification improvements can be calculated and visualized as a function of the construction parameters. The influence of different drive configurations can also be evaluated compared with the desired values.

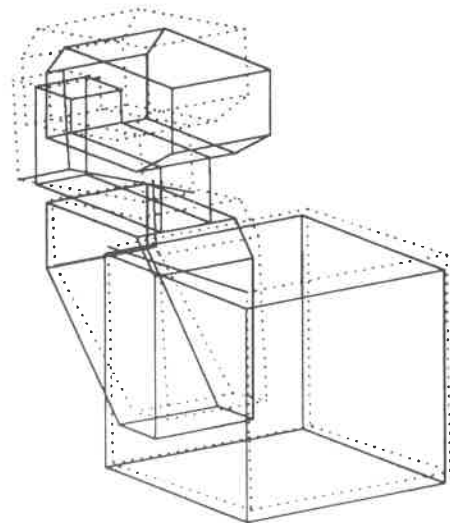


Figure 7: Animated mode-shape

Figure 7 shows an animated mode shape, as it can also be obtained from experimental modal analysis. The relative motion of tool and workpiece is immediately visualized, and modifications for improving the behavior can be directly verified.

Another aspect which becomes increasingly relevant are investigations concerning controllability and observability. These have to be applied in order to optimize the control and the structure together, which is necessary for achieving the maximum performance [5].

5 Application on parallel structures

This type of model can be used for serial structures, but is also valid for parallel structure machine tools as shown here. This approach is also useful for investigating the dynamic behavior of such structures. This leads to a better understanding of the behavior of these newly developed class of machine tools and allows better use of the potential inherent in these kinds of structures.

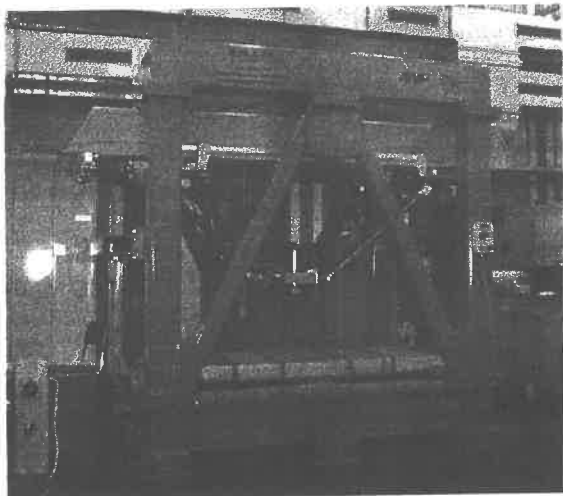


Figure 8: HEXAGLIDE test-bed

A representative example of parallel structure machine tools is the HEXAGLIDE, developed at the Swiss Federal Institute of machine tools and manufacturing. It employs six linear motors to orient a lightweight spindle assembly allowing six-degree of freedom movement of the tool with accelerations up to 2g. Its layout optimization was done using a rigid body model neglecting flexible modes.

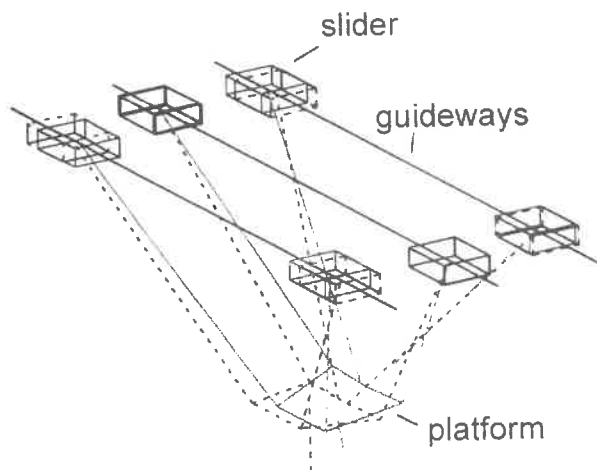


Figure 8: HEXAGLIDE mode shape (dotted: undeformed geometry)

Figure 8 shows the first rotation platform mode-shape. The rigid body lumped parameter approach led to a better interpretation of the results of experimental modal analysis that has been done on this structure. It also clearly showed the most efficient methods of improving the system's behavior [6].

6 Conclusion

The presented model of the machine tool in combination with the 2-D-measurement grid permits quick monitoring and optimization of machine tool performance. The exemplary investigations show the possibilities of pre-evaluating serial but also parallel machine tool structures. The wide range of illustrative results offer a variety of possibilities to minimize the re-work to be done, when the machine tool is already built. Virtual path measurements (simulations) can be performed. The approach presented here allows machine tool design to be optimized and the performance of the entire system to be checked and compared with the expectations in advance.

7 Acknowledgments

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Precision Control of Planar Magnetic Levitator

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We have developed a high-precision magnetic levitation stage for photolithography in semiconductor manufacturing [1,2]. This stage is the world's first maglev stage that provides fine six-degree-of-freedom (6-DOF) motion controls, and realizes large planar motions (50 mm × 50 mm) with only a single magnetically-levitated moving part, namely the platen. The key element of this stage is a linear motor capable of providing forces in both suspension and translation without contact. The stage operates with a positioning noise of as low as 5 nm rms in x and y , and acceleration capabilities in excess of 1 g (10 m/s²).

The magnetic levitator contains four three-phase linear permanent-magnet motors as labeled in Figure 1. Each linear motor can generate suspension (vertical) force as well as drive (lateral) force [1]. These four linear permanent-magnet motors collaboratively produce suspension forces to support the platen against gravity as well as drive it in six degrees of freedom. With an orthogonal arrangement of the motors as in Figure 1, the platen generates all 6-DOF motions for focusing and alignment as well as large two-dimensional step and scanning motions for a high-precision positioner as a wafer stepper stage in semiconductor manufacturing. For an elaboration of the motor operating principles, see [2].

In this extended abstract, we report recent test results with the planar magnetic levitator. These test results include system identification of the platen, fine settling performance, the effect of optical table oscillation, and the position ripple.

System Identification

To identify a linear model of the plant, we apply a stochastic system identification methodology that uses as random disturbance force input. The random disturbance is generated in software with the "rand()" function in the ANSI C language. The magnitude of the random disturbance is chosen to excite the plant persistently without losing stability. The frequency response of the plant shows a main structural resonance at about 750 Hz with no other significant resonance under that frequency. Comparing with the resonant frequency of the honeycomb sandwich platen at 1.2 kHz [1], the whole platen shows a lower resonant frequency. This is mainly due to the added masses of the four magnet arrays and the square mirror to the honeycomb sandwich. This high resonant frequency could be achieved with the planar magnetic levitator because of its simple mechanical structure, its low part count, and

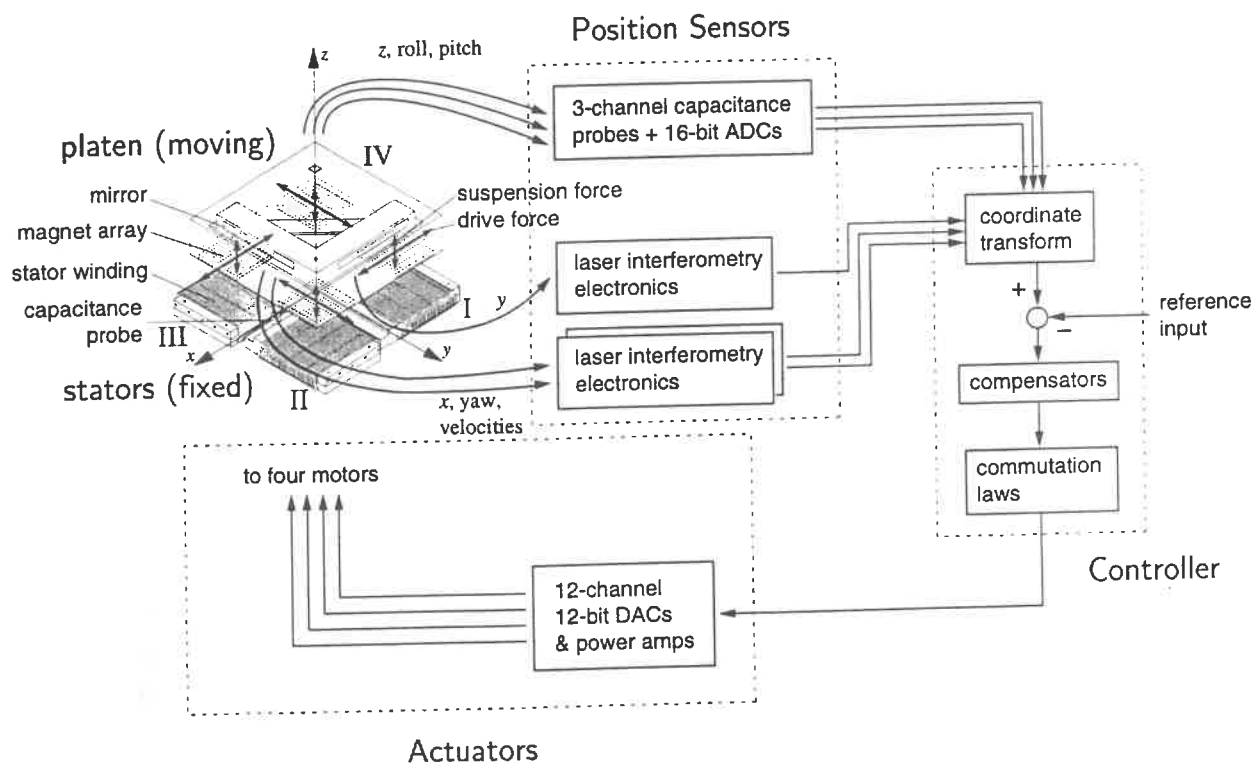


Figure 1: Control loop of the planar magnetic levitator

the high stiffness-to-mass ratio of the honeycomb construction of the platen. This high resonant frequency implies that a high-bandwidth control is possible with this one-moving-part structure.

Fine Settling

To demonstrate our stage's motion capability, we accelerate the platen at 1 g (10 m/s^2) until its velocity reaches the maximum slew rate of 200 mm/s of the laser head we use. The platen moves at this constant velocity for 80 ms and is decelerated at 1 g . This motion trajectory is typically referred to as a trapezoidal velocity profile. So, a 20-mm step command completes in 120 ms . The present lead-lag controllers need roughly 250 ms more to bring the errors in all six axes within their steady-state positioning noise level. This settling time can be reduced by implementing the control design. This can be done by: (1) Move the dominant closed-loop pole more deeply into the left half s -plane by sacrificing phase margin by moving the lag zero to the left in the s -plane. (2) Increase the whole closed-loop system bandwidth, which is limited by system structural resonances.

Optical Table Oscillation

Figure 2(a) shows a 40-mm step in x . The acceleration of the stage is set at $1/5\text{ g}$ (2 m/s^2) and the maximum velocity at 200 mm/s . It reaches the final positions in 0.3 s as commanded. Figure 2(b) is the final position error of the same step response on a finer scale. We observe a persistent, low-frequency (about 1.5 Hz), lightly-damped oscillation in the experimental

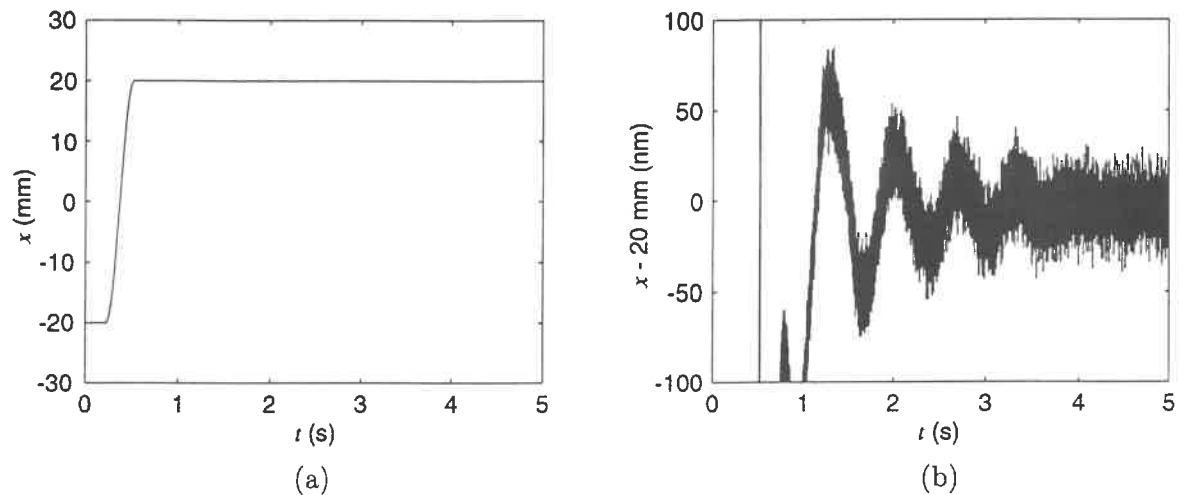


Figure 2: (a) 40-mm step in x . (b) Final position error of the same step response on a finer scale to show the effect of optical table oscillation

data. This is due to the oscillation of the optical table after completing the trajectory. This low-frequency oscillation is a genuine problem in any positioning system, whether based on magnetic levitation or an alternative technology, that requires fast response and high acceleration capabilities. This can be reduced with careful trajectory planning (with an S-shaped velocity profile, for instance) to avoid any abrupt acceleration change (jerk). Both an enhanced damping at the resonant frequency of the optical table and a high-mass vibration isolator will reduce the magnitude of this oscillation. For a more tight position window, it is recommended to use an active vibration isolation system. This reduces the oscillatory motions and rejects the floor vibration at the same time.

Position Ripple

Only the fundamental magnetic field components are usually taken into consideration to derive motor forces and commutation laws. Since it is impossible to implement perfect sinusoidal field sources in reality, there will be higher-order force components, so-called force ripple. This force ripple is different from cogging force which is prominent in slotted motors or variable reluctance motors, and is much smaller in magnitude. Similar, but bigger in magnitude, periodic error is also unavoidable in conventional lead screw systems due to the mechanical error of the screw and balls. These error motions can be reduced by open-loop correction. The present maglev system shows $\pm 1\text{-}\mu\text{rad}$ order position ripple in θ (i.e., the angle around the x -axis) during a commanded constant-velocity motion at -20-mm/s in the x -direction without any correction (Figure 3(b)). In a point-to-point positioning application as in a stepper, this force ripple has an insignificant effect on the overall system performance. There is no sinusoidal force ripple when the platen moves in the vertical direction since the vertical force depends on an exponential function of z . If the planar maglev stage had 6 phases instead of 3 phases, the ripple force would have been only 2% of the present ripple force [1].

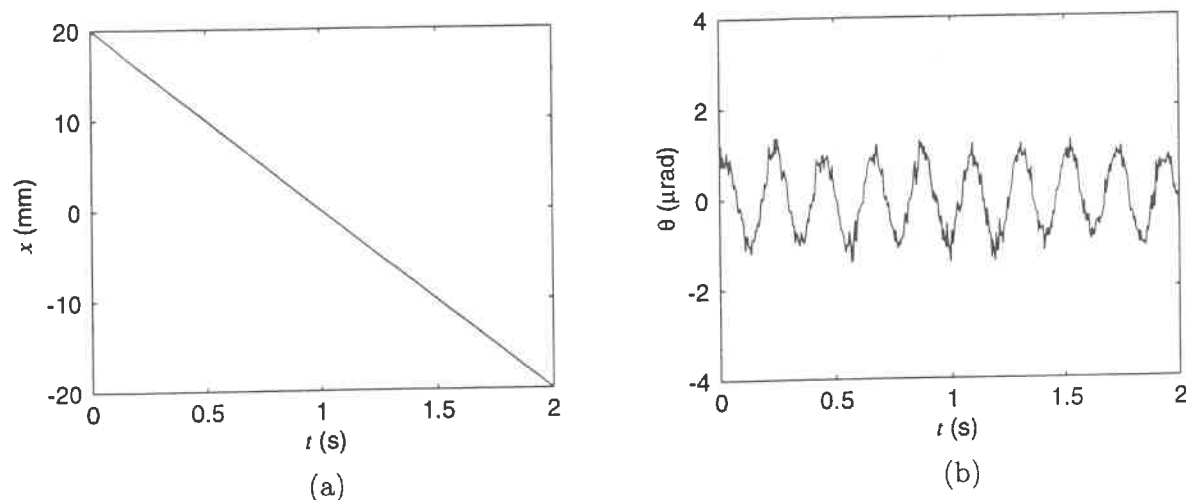


Figure 3: (a) -20-mm/s constant-velocity motion in x . (b) Position ripple in θ during the same motion

Follow-On Work

As follow-on work, we plan to demonstrate advanced performances of the planar magnetic levitator. Its performance enhancement is being pursued in a few ways:

(1) Increase the control bandwidth to get faster closed-loop system dynamics. Currently, the control loop could be closed at 150 Hz. The resonance of the platen at about 750 Hz, determined by our system identification result, is likely to be excited. We plan to implement notch filters at the platen resonant frequencies to suppress the vibration due to the resonance.

(2) Design better acceleration profiles and control laws to optimize the system response. The settling time can be reduced by control design by moving the dominant closed-loop pole more deeply into the left half s -plane by sacrificing phase margin by moving the lag zero to the left in the s -plane. Fine tuning of the controllers to optimize dynamic performance is another area for follow-on work.

(3) Correct error motions. We will seek for better modeling of the coupled dynamics with accurate estimations of the platen center of mass and rotational inertias required to minimize erroneous forces and torques in other axes. Besides error motions due to coupled dynamics, the current maglev system shows position ripple. This deterministic error motion due to force ripple can be reduced by feedforward correction.

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Experimental Studies on Magnetic Force Pre-Loaded Static Air Bearings

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A high stiffness and low cost air bearing on Magnetic Force Pre-Loaded Static Air Bearings (MLAB) was studied. Bearing performances were numerically simulated [1] by using Dover's Air Bearing Design Software. Several main-controlling parameters were included in the numerical simulation: air clearance, load capacity and stiffness. But discussions regarding numerical simulation will not be presented in this paper. This bearing used a magnetic force pre-load mechanism and Dover know-how DKH5 and DKH7 to achieve low cost and high stiffness performance. The relations among stiffness, air clearance, orifice size, weight effect and supply pressure were experimentally studied.

1. Introduction

Due to a very low viscosity and air compressibility air-bearings have lower stiffness than oil lubricated bearings [2], and due to high manufacturing costs, the increase of stiffness and decrease of cost become the goal of most designers. The efforts to increase stiffness as well as to decrease cost are the key points in our research. Opposite compensated air bearings have been shown either theoretically or practically that possess much higher stiffness than single side compensated bearings [3], but alternatively, opposite compensated bearings cost much more than single side compensated bearings.

In this study, the stiffness of a single side compensated air bearing, with C-channel magnetic force pre-loading [5] mechanism and Dover's "know how" technique for controlling pressure ratio, air

clearance as well as squeezing film damping, reached quite high values. The stiffness was an average of three times that in original design and the cost of the bearing was lower than original ones.

2. Experimental Preparations and Processes

Air bearing was rectangular shape 7.5 " x 16 " with 5 C-channel magnets that supplied total 265 lbs. pull force. Total twenty (24) orifices were arranged on both sides of magnets. Recess dimensions were 0.12 " x 0.22 " (Fig. 1). Testing equipment included, a Load cell pushed by a piston, ADE probes, LabView software and supporting frames (Fig. 2). In the tests, load cell was holding on piston that contacted to frame. Loading was applied on the center of the bearing.

Experiments were tested by changing (1) bearing supply pressure from 10 to 100 psi with increment of 5 or 10 psi, (2) weights of 30 lbs. and 123 lbs. on bearing, (3) diameter of orifices of 0.002 " and 0.003 " and (4) Dover's Know How (DKH) with DKH5 and DKH7. The loading range through load cell applied on bearing was from 0 to 10 lbs.

3. Experimental Results

Bearing stiffnesses and air clearance vs. bearing supply pressure with different weights were shown in Fig. (3). The results on this figure were come from without DKH being used.

There exists a maximum stiffness with respect to supply pressure. In our studies, for 0.002 " orifices, the maximum stiffnesses were at 20 psi when adding 123 lbs. and at 15 psi when adding 30

lbs. For 0.003 " orifices, the maximum stiffnesses were at 25 psi when adding 123 lbs. and at 20 psi when adding 30 lbs. Over the entire pressure range of 10 to 100 psi, stiffness was increased with larger weight on bearing, regardless of orifice size and weights. The stiffness of bearing with 0.002 " orifices was always larger than that with 0.003 " orifices with respect to same weight on bearing. The maximum stiffness was shifted to a lower pressure when orifices were change to 0.003 " from 0.002 " .

Regarding air clearance, the studies showed that the air clearance was always increased with the increase of supply pressure regardless of orifice size and weight. Lighter weight on bearing has larger clearance than heavier weight. The on all air clearance curves always happens. On this bearing, the inflexion happened at 20 psi when using 0.003 " orifices and at 13 psi when using 0.002 " orifices.

A comparison between stiffnesses without DKH and with DKH was shown on Table 1. The stiffness was significantly increased by using DKH, and DKH5 applied more function than DKH7.

4. Discussions

The inflexion on air clearance curves on Fig. 3 was due to air choke in orifice when pressure ratio was decreased lower than 0.53. Choke point was a critical design point for maximum stiffness.

Stiffnesses in MLAB possessed a maximum value with respect to supply pressure. The supply pressure for maximum stiffness will shift to higher pressure when putting more weight on bearing. The stiffness was decreased with an increase of supply pressure. There were two reasons for this: one was due to larger air clearance, another one was due to decreased compensation caused by choke at orifices. The pressure ratio of recess pressure to supply

pressure was only 0.2 @ 80psi, 0.26 @ 60 psi, 0.36 @ 40 psi. When supply pressure decreased to 20 psi, the pressure ratio was 0.56 that was larger than critical ratio of 0.53. But when pressure decreased again, the stiffness started to drop. When supply pressure decreased from 100 psi to 10 psi, air clearance was decreased, which was beneficial to increase stiffness. For highest stiffness, the optimum ratio in MLAB design was 0.53 ~ 0.56. In other words, the maximum stiffness happened at the critical pressure ratio. For all testing conditions, such as with more weight, with different orifices, the same result were always achieved.

Weights on MLAB had a function on increasing stiffness. It is very obvious that when weight was increased, the clearance was decreased, which resulted in an increase in stiffness.

Another result showed a relation between stiffness and orifices size. Through the tests, larger orifices will lead to a decrease in stiffness, which was mainly caused by higher efficiency compensation of smaller orifices. Authors did not find over-compensation of orifices when using smaller orifices.

The comparison of stiffnesses of MLAB with DKH and without DKH showed up on Table 1. When using 0.002 " orifice, applying 123-lbs. weight on bearing and supplying pressure 80 psi, vertical stiffness was increased from 1,308,000 lb./in. to 1,781,000 lb./in. by using DKH7 and 2,036,000 lb./in. by using DKH5. The maximum stiffness was a 35% increase.

5. Conclusions

5.1 For magnetic force pre-loaded air bearings, there was an optimum supply pressure to achieve the highest stiffness. In this study, the optimum supply pressure was found at the point when pressure ratio

reached a critical ratio. Higher or lower than this value will lead to a decrease in stiffness.

5.2 More loading on bearing will result in higher stiffness and supply pressure for maximum stiffness will shift to higher pressure with the increase of weight on bearing.

5.3 Smaller orifices will result in higher stiffness than larger orifices.

5.4 The optimum supply pressure will be shifted to higher pressure when orifice was increased.

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Table. 1* Vertical stiffness (x 1,000 lb./in.)

pressure (psi)	30 lbs weight	123 lbs weight	123 lbs weight (DKH7)	123 lbs weight (DKH5)
40	1100	1580		
50	1034	1486		2000
60	928	1395		2018
80	894	1308	1781	2036
100				1927

* orifice size 0.002 ". Weights were applied on bearing. DKH is Dover Know-How level.

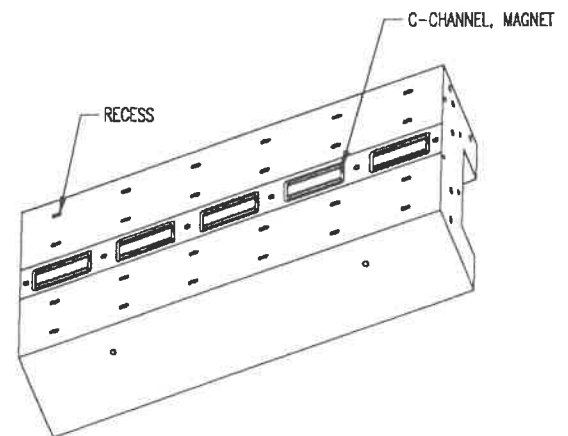


Fig. 1 Structure of air bearing

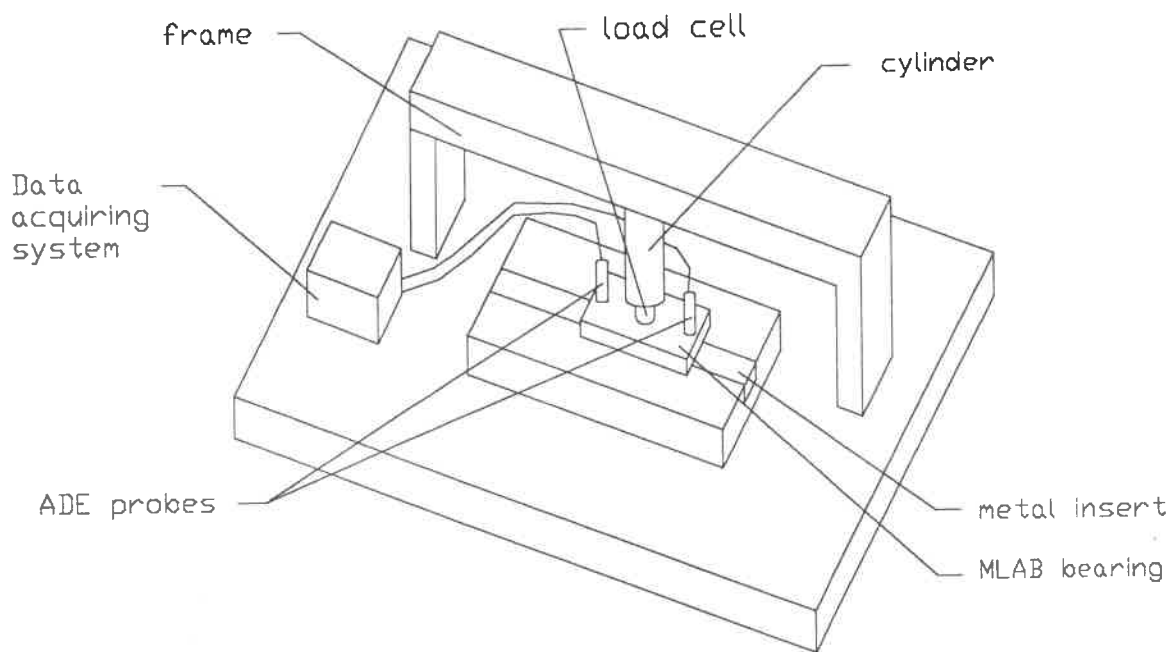


Fig. 2 Experimental Setup

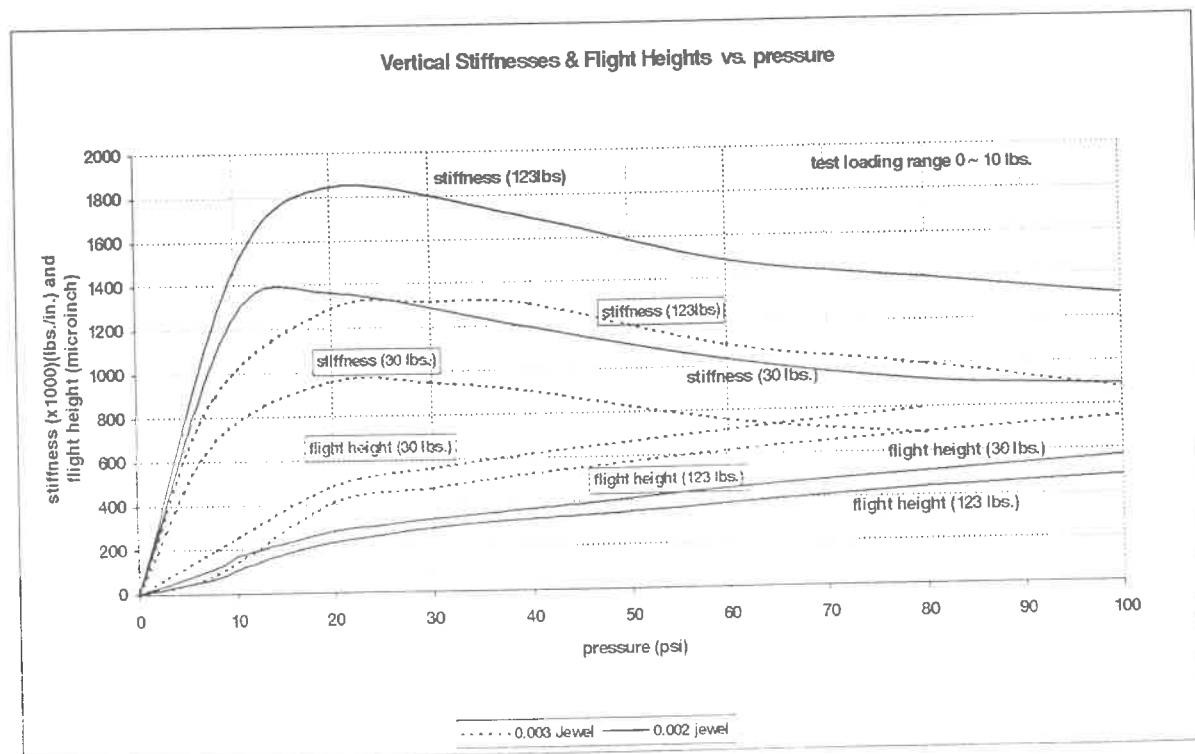


Fig. 3 Bearing stiffnesses and air clearance vs. bearing supply pressure with different weights on bearing without DKH

A Magnetically Guided Precision Drive with 6 Degrees of Freedom

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Abstract

A new magnetically guided precision drive is presented. The force generation and measurement systems for both the drive and the bearing are described. The paper closes with the introduction of the hybrid control concept.

Introduction

Through the application of direct drives a significant advance in the field of precision drive technology could be achieved during the past years. The bearing of the armature remains, however, a problem, particularly for applications in clean rooms or vacuum. Mechanical bearings limit the achievable accuracy and speed. Their wear leads to contamination and reduces the lifetime. A good solution to the problem are aerostatic bearing elements that are used in the multi-coordinate drive developed at the Technical University of Ilmenau (Figure 1, left side).

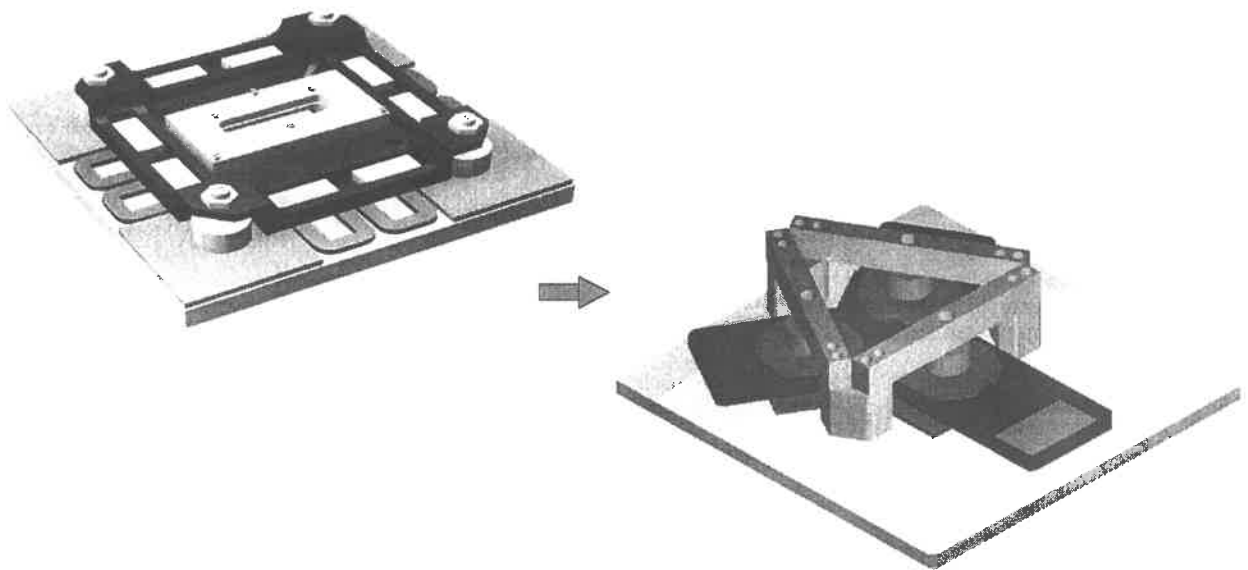


Figure 1. Trend in the development of planar drives

However, aerostatic bearing elements are not suitable for clean-room and vacuum applications. Active magnetic bearings do not possess these disadvantages (Figure 1, right side). This is, however, not their only benefit. More important is the position control in the degrees of freedom that are restricted by conventional bearings. Such solutions were described in [1] and [2].

Design

Figure 2 gives an overview over the structure of the drive:

The stage (6) is magnetically supported by three pot magnets (2) that exert forces on iron plates (3) mounted on the stage. The stage also carries the NdFeB magnets (4) whose magnetic circuits are closed by the iron yokes (12) and again the plates (3).

The flat drive coils (11) are arranged in a triangle. The drive force is produced by all three elements. A rotation of the stage is also possible in this setup.

The stage further carries a cross grid (5) for the incremental measuring system (10) that is mounted on the stator (7). The measuring system determines the position of the stage in x , y and ϕ and tolerates a certain motion along z .

The iron plates (8) mounted on the stage serve as targets for the z position measurement with three eddy-current sensors (9) that are fixed on the stator.

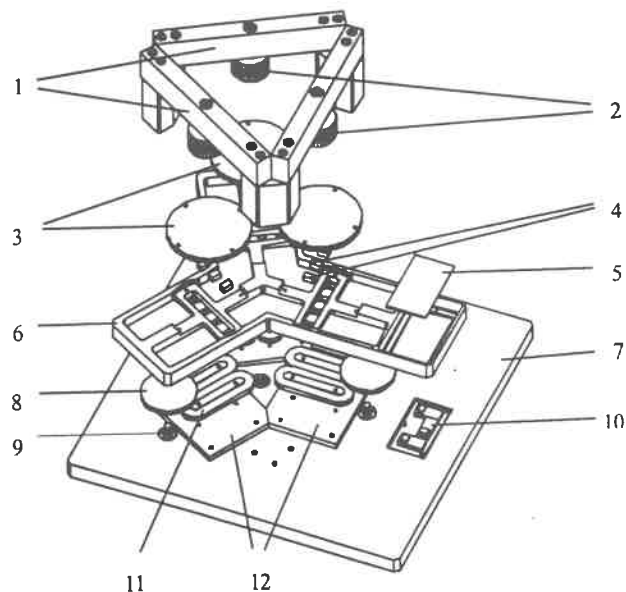


Figure 2. Explosive view of the planar drive

Force generation

The drive combines three planar direct drive elements with three magnetic bearing elements. The direct drive elements utilize the electrodynamic force generation principle with moving magnets. This allows for a cable-free stage design and, thus, avoids drag forces. The attraction of the magnets to the stator implements part of the force for the magnetic bearing which, therefore, needs just three carrying magnets. The triangular arrangement of the elements allows to generate forces in x and y as well as a moment around z . For the generation of the drive forces three of the elements shown in figure 3 are used (see [1]). As the stage moves the overlap of single coil and magnet pairs changes leading to changes in the force generated. For the whole arrangement, however, the generated force remains nearly constant. Electronic commutation and in-situ calibration ensures that the drive force stays within $\pm 1\%$ margins for the whole range of travel.

The magnetic bearing supports an area of motion of $30 \times 30 \text{ mm}^2$ in the x - y -plane. The weight of the stage and the attraction forces between the permanent magnets in the stage and the stator are compensated by the bearing.

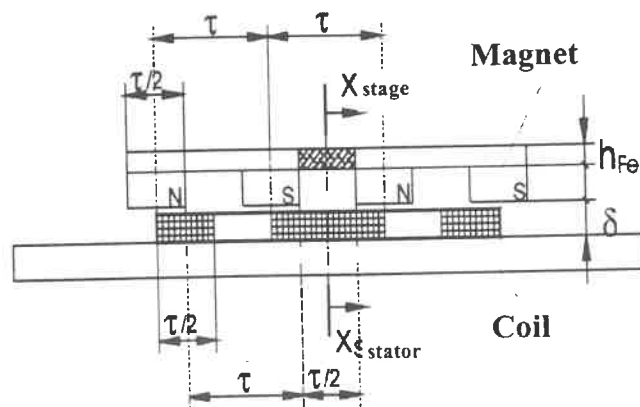


Figure 3. Structure of the force generation elements

Measurement system

An incremental incident light measuring system with a cross grid measure and software interpolation allows to determine the stage position in the drive coordinates x , y and φ with a resolution of 20nm or 0.1" respectively. The value of φ is restricted to $\pm 2^\circ$. The stage position in the bearing coordinates z , r_x and r_y is measured using three eddy-current sensors with a resolution of 0.1 μ m or 0.1" respectively. The accuracy of the measurement systems significantly determines the parameters of the whole drive.

Control

The control of such a complex system requires a powerful hardware. Figure 4 shows the hardware structure of the control system based on a DSP system from dSPACE.

Three hybrid controllers are used for each of the magnetic bearing elements [2]. In the hybrid scheme an analogue PDD²-controller is supervised digitally (see figure 5). D² signifies a double-differentiating term. The supervision tunes the analogue controller depending on the stage position and the load. It also implement integral action to achieve zero steady-state error. The hybrid scheme ensures a high stability of the bearing parameters over the whole operating range. It also compensates the existing coupling and nonlinearity. This enables both a high dynamic stiffness, due to the continuous control, and a very high static stiffness witch depends only on the resolution of the measurement systems. Figure 6 shows the position noise of the magnetic bearing mainly due to the eddy-current sensors. Position accuracy of 1 μ m and a stiffness of 1 N/ μ m within a ± 0.5 mm range of motion was achieved. Using better sensors can improve the bearing parameters significantly.

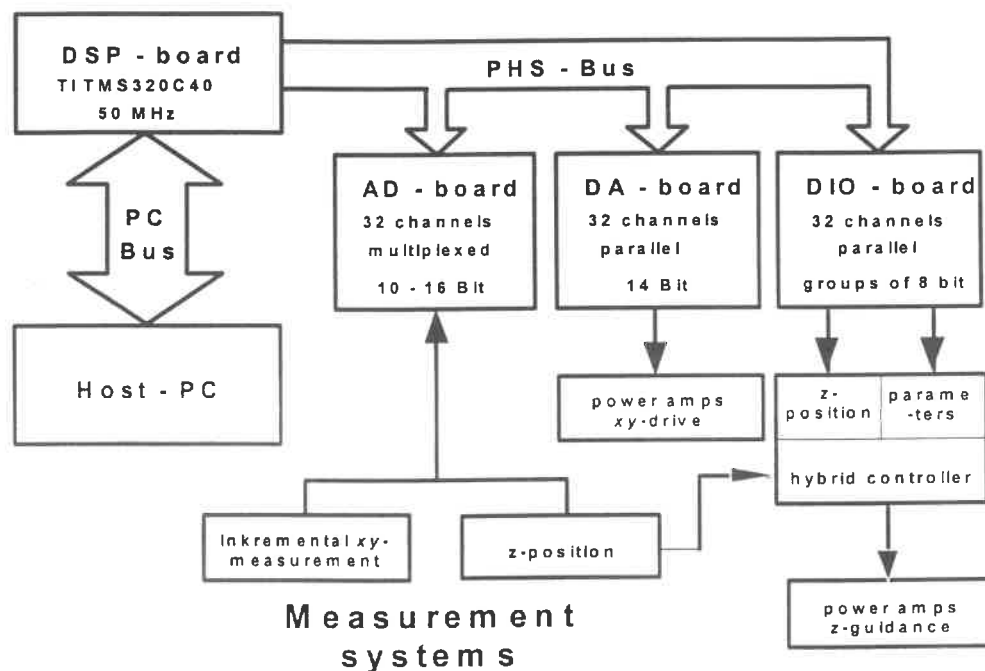


Figure 4. Structure of the control system hardware

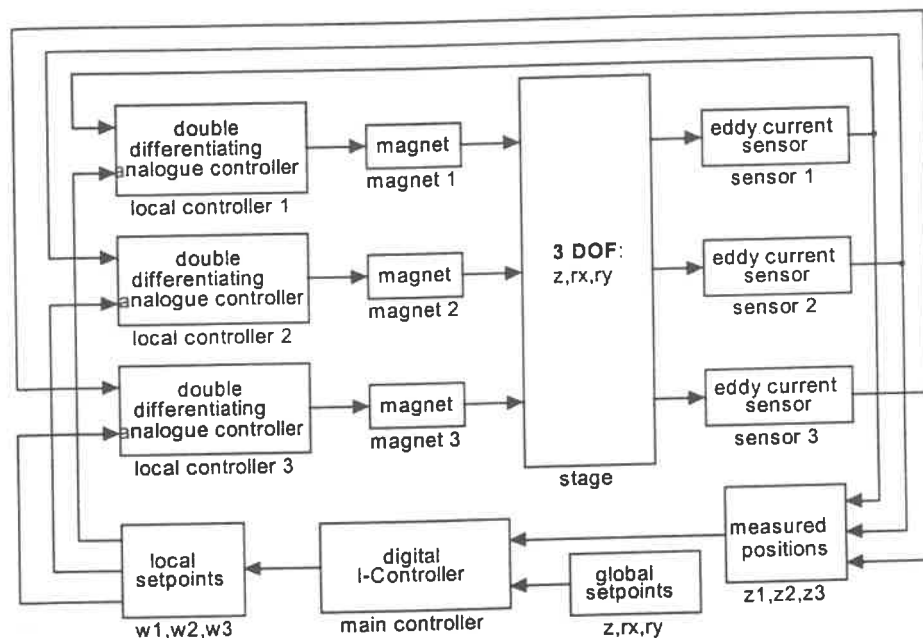


Figure 5. Structure of the magnetic bearing control

combines the advantages of state-feedback control with the robustness and static disturbance cancellation properties of classical PID-control. It implements a special time-discrete, linear-quadratic control with state estimation based on an incremental system model which eliminates steady-state control errors via its integral part taking also limits of the control action into account [4]. An accuracy of $0.2 \mu\text{m}$ was achieved for the developed drive within its $30 \times 30 \times 0.2 \text{ mm}^3$ range of motion.

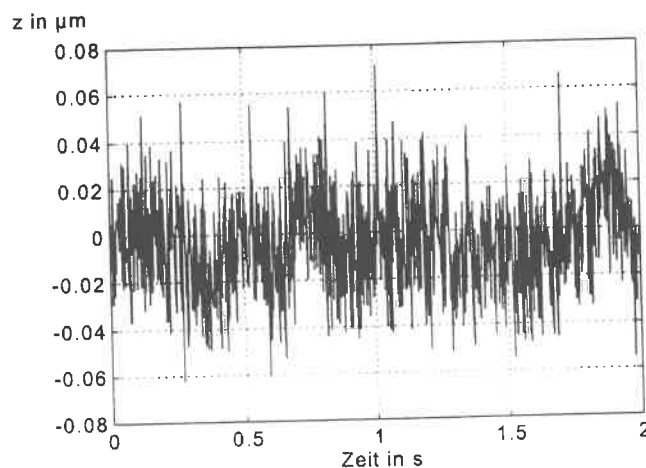


Figure 6. Position noise of the magnetic bearing.

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Effect of Modulation on Single Grit Scratch Tests

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1 Introduction

Precision ceramic components are being used in an increasing variety of engineering applications. Compared to metals, ceramics are much harder and more susceptible to brittle fracture. Ceramic components, in most cases, require finishing operations to achieve surfaces of required geometry, tolerance, and finish. Grinding is an enabling technology in the manufacture of ceramic components because it produces accurate shapes deterministically. But grinding generally induces surface and sub-surface damage in the finished part, which can severely degrade its strength and life in service conditions. The grinding of fracture free surfaces is possible, but it requires removing material in very small chips which limits material removal rates (MRR). Reducing depth of sub-surface damage, even if it cannot be eliminated, is favorable in terms of part strength and life. Small amounts of damage require only small amounts of secondary processing such as lapping. While secondary processes eliminate sub-surface damage, they increase cost and make figure accuracy worse.

To increase MRR, several researchers have taken advantage of ultrasonically assisted machining [1,2]. In an attempt to understand the mechanism behind the beneficial effect of this method, we have previously investigated the effect of high frequency modulation on chip geometry and cutting force [3,4]. We learned that at the high cutting speeds typical of grinding, modulation does not significantly reduce the chip size or frictional force. In this paper, we focus on its potential to reduce subsurface damage.

2 Theoretical Motivation

Grinding produces damage normal to the finished surface (median and radial cracking) and parallel to the finished surface (lateral cracking). The normal damage is typically left behind in the finished part and is responsible for its strength degradation. The lateral damage, on the other hand, facilitates material removal [5] thus enhancing MRR.

Normal damage nucleates and propagates under loading. Lateral damage nucleates and propagates upon unloading. In a conventional grinding process, loading always precedes unloading. As force/grit increases gradually from zero, normal damage initiates first and propagates to its maximum depth at peak load. Only later, upon unloading, does the lateral damage responsible for material removal initiate. Thus, in a traditional grinding operation on brittle materials, detrimental normal damage is usually fully developed before the material removal mechanism of lateral damage has had a chance to initiate. Significant improvements to the grinding process could be realized by: (1) initiating the material removal mechanism of lateral damage early in the process

before normal damage is fully developed; (2) utilizing the shielding effects of lateral damage to retard and potentially eliminate normal damage from the finished product.

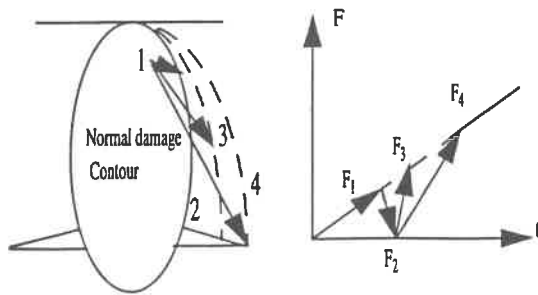


Figure 1: Illustration of the shielding effect

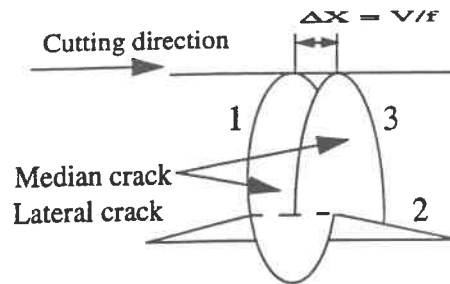


Figure 2: Shielding effect in single point modulation cutting

The shielding effect is shown schematically in Figure 1. First, a load F_1 is applied and produces median crack 1. Unloading to $F_2 = 0$ initiates the lateral crack 2. Then a load F_3 (bigger than F_1) is applied and produces the median crack 3. The lateral crack acts as a barrier against further penetration of normal cracks upon reloading. The median crack 3 cannot go beneath the lateral crack 2. However, a load larger than F_4 would produce a median crack that is big enough to pass the lateral barrier and go deeper. Theoretical calculation shows that for indentation loads between 0.1 N and 300 N, $F_4 = 3 \cdot F_1$ [6].

Modulation can help to achieve the shielding effect in grinding. The difference between the indentation event (Figure 1) and modulation cutting (Figure 2) is that for Figure 1, the workpiece is indented and reindented at the same point, while in Figure 2, the workpiece is indented at one point and reindented at a different point. Figure 2 shows that if Δx is not too big, the lateral crack can act as a barrier against further penetration. Vertical modulation moves the cutting grit (or workpiece) in a sinusoidal wave that increases and decreases the cutting force. When the cutting grit first contacts the workpiece, the median crack 1 is produced as shown in Figure 2. After the grit unloads, the lateral crack 2 initiates. Reloading the workpiece produces the median crack 3. The lateral crack 2 acts as a barrier against further penetration of the normal damage thus achieving the shielding effect. The distance between the two adjacent points of maximum force is $\Delta x = V/f$ where V is the cutting speed and f the modulation frequency. Reducing Δx (by decreasing V or increasing f) should increase the shielding effect and reduce the depth of normal damage.

3 Scratch Experiments

To test the concept of using modulation to promote crack shielding, we performed single grit scratch experiments. Figure 3 shows the experimental setup. The workpiece is pyrex glass and excited to vibrate in the Z direction by a magnetostrictive actuator. A diamond cutting nib is mounted on the spindle. Underneath the cutting nib is a force sensor. The table is moved in the Z direction until the reading of the force sensor is 0.1 N. At this point, we assume the cutting nib is just in contact with the workpiece and has zero depth of cut. To make a scratch, the actuator extends, modulates, and retracts within one spindle rotation.

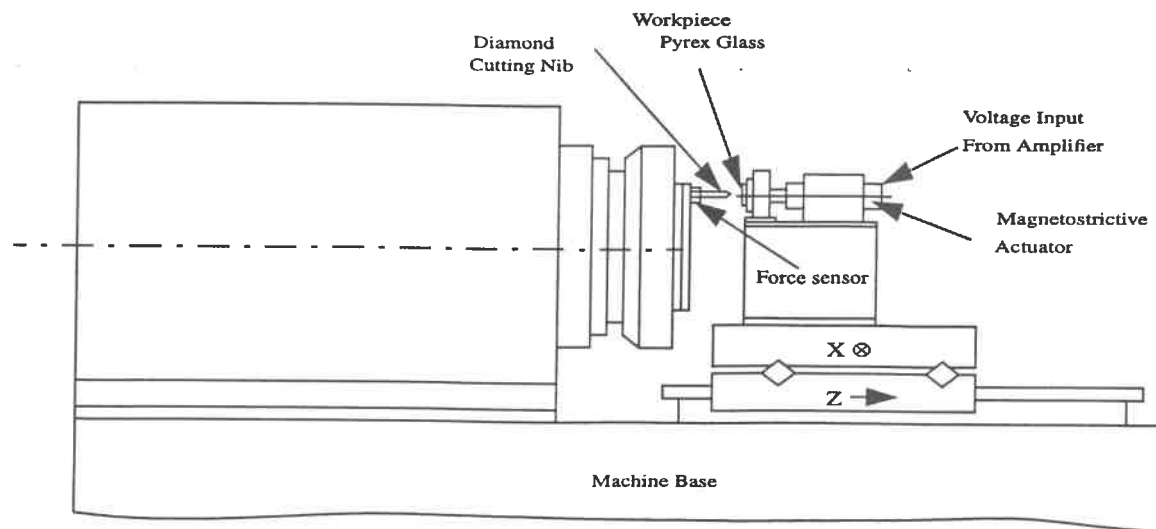


Figure 3: Schematic illustration of the lathe with the actuator

To permit measurement of the radial depth of damage, we prepare workpieces as shown in Figure 4. Two glass sections are glued together, and their top surface is polished flat. After making scratches on the polished surface, we separate the combined workpiece to view the radial cracks and measure their length.

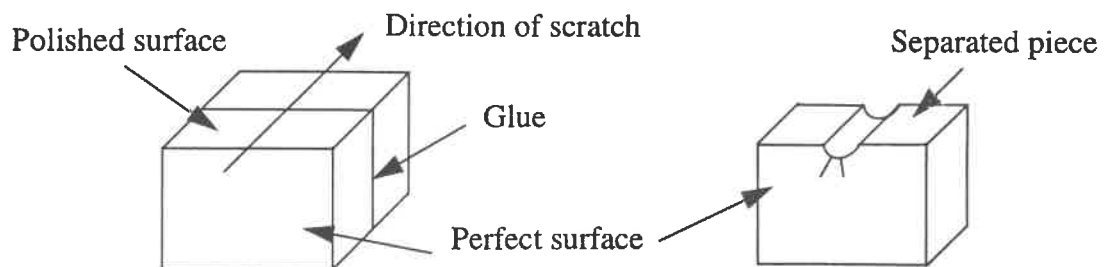


Figure 4: Illustration to show how to evaluate the radial depth of cut

Figure 5 shows a top view of scratches made without and with modulation. The circular regions on the lower scratch occur at the low point in the modulation cycle and indicate broad lateral cracks. When the lateral cracks do not overlap, as in this case, there is no shielding effect. A higher modulation frequency or lower cutting speed would achieve overlap and shielding.



Figure 5: Top view of scratches without modulation and with modulation (1kHz, DC=10 μm , Ampl=10 μm , cutting speed =17 cm/s) (picture width=1.1 mm)

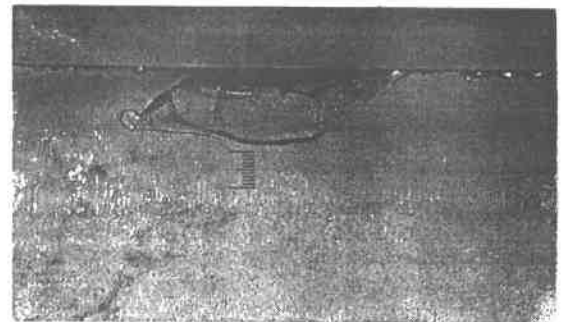


Figure 6: Front view of scratch with modulation (1kHz, DC=10 μm , Ampl=10 μm , cutting speed=8.6 cm/s) (marker size=100 μm)

Figure 6 shows a front view of a scratch produced at a lower cutting speed. A lateral crack is evident at the bottom of the damage zone. No radial cracks extend below it, which suggests that shielding occurred. The average radial damage depth for this scratch condition was $176\text{ }\mu\text{m}$. For the same total depth of cut ($20\text{ }\mu\text{m}$) without modulation, the average radial damage was $231\text{ }\mu\text{m}$.

We performed experiments over a range of modulation frequencies. Figure 7 shows that the depth of radial crack reduces as modulation frequency increases. This is consistent with the earlier discussion on the role of Δx (see Figure 2).

4 Conclusions and Future Work

The single grit scratch experiments validate the concept of using modulation to promote crack shielding and reduce sub-surface damage depth. A low ratio of cutting speed to modulation frequency is most beneficial. These scratch tests were performed at low cutting speeds; future work will include testing at higher speeds and modulation frequencies. In addition, more work is needed to find the optimum modulation amplitude and frequency/speed ratio. We also plan to assess the effect of modulation direction. Finally, grinding tests will be performed. It is unclear how the scratch results will extend to grinding where factors such as crack interactions and chip loading will play a role.

Acknowledgments

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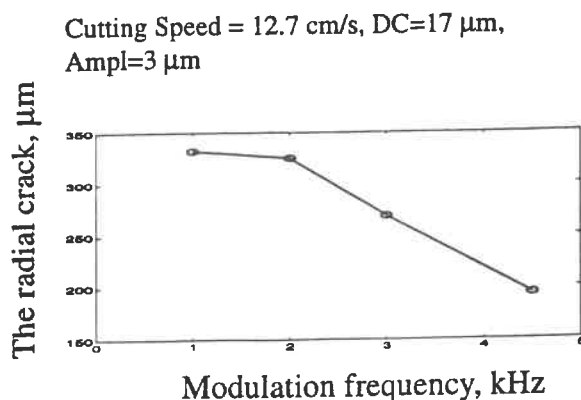


Figure 7: Frequency effect on radial cracks

Vitreous Bond CBN High Speed and High Material Removal Rate Grinding of Ceramics

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Abstract

High speed (up to 127 m/s) and high material removal rate (up to $10 \text{ mm}^3/\text{s}/\text{mm}$) grinding experiments using a vitreous bond CBN wheel were conducted to investigate the effects of material removal rate, wheel speed, dwell time and truing speed ratio on cylindrical grinding of silicon nitride and zirconia. Experimental results show that the high grinding wheel surface speed can reduce the effective chip thickness, lower grinding forces, enable high material removal rate grinding and achieve a higher *G-ratio*. The radial feed rate was increased to as high as $0.34 \text{ } \mu\text{m}/\text{s}$ for zirconia and $0.25 \text{ } \mu\text{m}/\text{s}$ for silicon nitride grinding to explore the advantage of using high wheel speed for cost-effective high material removal rate grinding of ceramics.

1. Introduction

The high speed cylindrical grinding experiments presented by Shih, et al. (1998) are extended to study the CBN grinding of silicon nitride and to explore the high speed, high material removal rate grinding of zirconia and silicon nitride. The truing process which generates the lowest grinding forces was used in the high material removal rate grinding of zirconia and silicon nitride. The effect of dwell time was investigated to identify the improvement in surface finish and roundness.

Silicon nitride and zirconia have been recognized as difficult to machine materials. The CBN abrasive was selected because it can be trued to the μm -level precision form by a rotary diamond wheel and it has the flexibility to grind both ceramics and hardened steels. Table 1 compares the properties of the three work-materials, silicon nitride, zirconia and hardened M2 steel, the three work-materials used in this and previous high speed grinding tests (Shih et al., 1998). It can be seen that silicon nitride has significant higher hardness and elastic modulus than the other two materials. The ceramic grindability test method developed by Chand and Guo (1996) was used to quantify the grindability of zirconia and silicon nitride. Test results showed that the grindability index, G_c , for zirconia and silicon nitride were 31.9×10^{-4} and $10.6 \times 10^{-4} \text{ mm}^3/\text{Nm}$, respectively. Comparing to the G_c of other ceramics, these two work-materials have relatively low G_c , indicating that both materials are difficult to grind.

Table 1. Comparison of material properties of the zirconia, silicon nitride, and hardened M2 steel.

	Zirconia	Si ₃ N ₄	M2 Steel
Manufacturer	Coors	Toshiba TSN-10	Latrobe
Grain size, μm	40-60	1-2	22
Density, g/cm^3	5.6	3.3	7.8
Hardness, Knoop 50g	1000	1700	860
Bending flexural strength, MPa	620 ~ 690	700 ~ 850	4300
Elastic Modulus, GPa	200	310	200
Fracture Toughness K_{IC} , $\text{MPa}\cdot\text{m}^{1/2}$	11	6 ~ 7	7.6
G_c , Grindability Index $\times 10^{-4} \text{ mm}^3/\text{Nm}$	31.9	10.6	N/A

N/A: Not applicable

The experiments were conducted on a Weldon AGN5 CNC cylindrical grinding machine. The grinding wheel was slowed down during truing and increased to the desired speed during grinding. This technique has been proven useful to reduce the grinding forces. Detailed process parameters have been documented in Shih, et al. (1998). Some critical parameters were: coolant flow rate was $62.5 \text{ l}/\text{min}$, workhead speed was 600 rpm , width of plunge grinding was 6.35 mm and grinding wheel was the Cincinnati Milacron 2BN 320-Q164-VMB. The next two sections summarize the results on high speed grinding of silicon nitride and high material removal rate grinding of both silicon nitride and zirconia.

Table 2. Selection of the sixteen grinding tests.

v_t (m/s)	v_s (m/s)				q_d
	35	61	91	127	
19.3	x	x	x	x	0.661
25.4	x	x	x	x	0.503
33.9	x	x	x	x	0.381
35	x				0.364
61		x			0.209
91			x		0.140
127				x	0.101

x: Selected for the test

2. High speed grinding of silicon nitride

The sixteen grinding tests (Shih et al., 1998), as listed in Table 2, were used to test the effects of two setup parameters, grinding wheel surface speed (v_s) and truing speed ratio (q_d), on silicon nitride grinding. Truing speed ratio is defined as the ratio of the truing wheel surface speed (v_t) to v_s , i.e., $q_d = v_t/v_s$, is the surface speed of the truing disk and $q_d = v_t/v_s$. Figure 1 shows the results of grinding tests for silicon nitride.

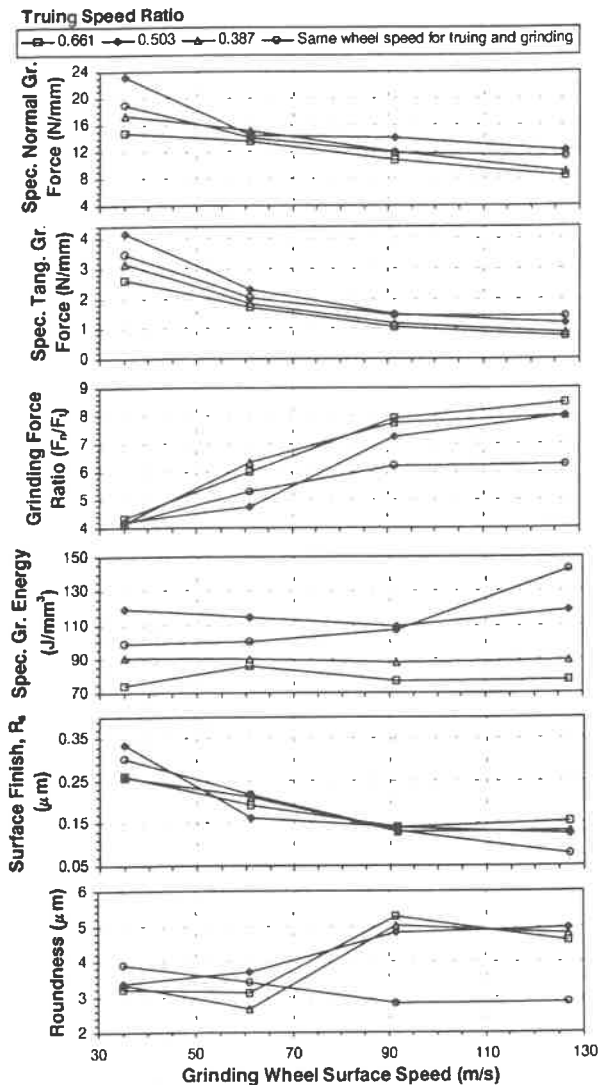


Fig. 1. Specific grinding forces, grinding force ratio, specific grinding energy, surface finish, and roundness of CBN grinding of silicon nitride.

2.1. Specific normal and tangential grinding forces, f_n and f_t , and grinding force ratio

Note that lowercase letters are used to distinguish specific force (force per unit length) from force, which is shown in uppercase. Compared to the grinding results for zirconia and M2 (Shih, et al., 1998) using the same machine and CBN grinding wheel, f_n and f_t are much higher and F_n/F_t is about the same for grinding silicon nitride.

Effect of v_w Similar to the trend observed before, lower grinding forces and a higher grinding force ratio were generated at higher v_w . When grinding silicon nitride, the lowest f_n and f_t could be achieved at high v_w ($=127$ m/s) and high q_d ($=0.661$). They are 8 and 0.8 N/mm, respectively, which are very high compared to the values grinding zirconia and M2 steel. Such high grinding forces will adversely affect the wheel wear rate.

F_n/F_t , the normal-to-tangential force ratio, represents the tribological behavior between the grinding wheel and workpiece. F_n/F_t remains low, below 10, for silicon nitride grinding at high coolant flow rate. This is due to the relatively high F_t in silicon nitride grinding. The trend is similar for zirconia grinding. F_n/F_t increases at higher v_w , i.e., the F_t decreases at a faster rate than F_n or the friction coefficient decreases at high v_w .

Effect of q_d Truing at the highest q_d ($=0.661$) is beneficial because grinding forces are reduced. However, unlike the four distinctly different levels of grinding forces at different q_d for zirconia and M2 steel, the overall effect of q_d on f_n , f_t , and F_n/F_t is not obvious for silicon nitride grinding.

2.2. Specific Grinding Energy, u

Silicon nitride sublimates at about $1900^\circ C$. The specific energy required to transform silicon nitride from the solid state at room temperature to the gaseous state at $1900^\circ C$ is only $4.2 J/mm^3$. The calculated u for silicon nitride grinding obtained from this experiment was much higher, exceeds $70 J/mm^3$. It is also higher than the lowest calculated u for zirconia and M2 grinding, which equals 11.6 and $18.9 J/mm^3$, respectively.

Effect of v_w The high v_w does not result lower u . This is different from the trend in zirconia grinding.

Effect of q_d Truing at the highest q_d ($=0.661$) yields the lowest u during grinding. However, the overall effect of q_d on u is not as clear as in the zirconia and M2 steel grinding.

2.3. Surface finish, R_a

Effect of v_w A trend of improving surface finish (lower R_a) at higher v_w could be seen for silicon nitride grinding, which is different from the zirconia and M2 steel grinding. At $v_w=127$ m/s, the R_a of ground parts were all better than $0.15 \mu m$.

Effect of q_d The effect of q_d on surface finish was not significant. The four plots of surface finish at different q_d were roughly parallel.

2.4. Roundness

Unlike the trend observed for surface finish, higher v_w does not always generate better roundness. The roundness at $v_w=127$ m/s and $q_d=0.661$ is quite high ($4.6 \mu m$). The effect of q_d on roundness cannot be seen.

3. High Material Removal Rate Grinding of Zirconia and Silicon Nitride

At high speed, more aggressive feed rates can be applied to achieve high material removal rate, reduce grinding time and possibly create savings in

Table 3. Radial feed rates used in this study and their corresponding grinding time, specific material removal rate, material removal per revolution, and equivalent chip thickness.

	Radial feed Rate, $\mu\text{m}/\text{s}$	0.085	0.13	0.17	0.25	0.34
Grinding time (s): 5.613 mm (0.221 inch) diametrical stock removal		31.7	21.1	15.8	10.6	7.91
Specific Material Removal Rate ($\text{mm}^3/\text{s}/\text{mm}$): Zirconia (9 mm dia.)		2.39	3.59	4.79	7.18	9.58
Silicon nitride (9.525 mm dia.)		2.53	3.80	5.07	7.60	10.13
Material removal per revolution, a (μm)		8.47	12.7	16.9	25.4	33.9
Equivalent chip thickness, $h_{\max}$ (μm) $v_w=35$ m/s		1.67	1.85	1.99	2.20	2.36
(for 9.525 mm dia. silicon nitride at start of the grinding) $v_w=61$ m/s		1.27	1.40	1.50	1.67	1.79
$v_w=91$ m/s		1.04	1.15	1.23	1.36	1.47
$v_w=127$ m/s		0.88	0.97	1.04	1.15	1.24

production. One of the objectives in this study is to investigate the effects of feed rate and dwell time on the high speed grinding of ceramics. Three setup parameters in this test were: feed rate, dwell time and work-material. As shown in Table 3, five feed rates were tested. When grinding the silicon nitride at the highest radial feed rate, $0.34 \mu\text{m/s}$, the high grinding forces stalled the workpiece due to the lack of power in the chuck motor. Tests for silicon nitride grinding at such high feed rate were abandoned. Two levels of dwell time, 3 and 10 s, were added to study its effects on surface finish and roundness. In summary, a total of 18 tests, 10 for zirconia and 8 for silicon nitride, were carried out.

At high grinding wheel surface speed, the equivalent chip thickness can be estimated using the following formula (Mayer and Fang, 1997).

$$h_{max} = \sqrt{\frac{4v_p}{v_w Cr}} \sqrt{\frac{a}{d_e}}$$

where

$d_e = d_w / (1 + d_w / d_p)$ for external grinding, d_p is the diameter of the workpiece and d_w (=406 mm) is the diameter of the grinding wheel. C (=38 grits/mm²) is the number of active cutting points per unit area on the wheel surface. v_p is the surface speed of the workpiece. r (=10) (Mayer and Fang, 1997) is the ratio of chip width to average undeformed chip thickness for ceramic grinding.

The material removal per revolution and equivalent chip thickness, h_{max} , at four different feed rates are listed in Table 3. The equivalent chip thickness at 127 m/s wheel speed is about half of that at 35 m/s wheel speed. Table 3 also shows the grinding time for a 5.613 mm diametrical material. It takes only 7.91 sec at 0.34 $\mu\text{m/s}$ radial feed rate to grind the zirconia part from 9 to 3.387 mm diameter. Due to the difference in part diameters (9 vs. 9.525 mm), the specific material removal rates are slightly different for zirconia and silicon nitride.

In the high feed rate tests, $v_w=127$ m/s and $q_a=0.661$, which generate the lowest grinding, were used. Figure 2 shows the effects of feed rate, dwell time and work-

material on grinding forces (f_n and f_t), grinding force ratio (F_n/F_t) and specific grinding energy, u . The change of dwell time changes the surface finish and roundness, which are illustrated in Fig. 3. The linear regression analysis was applied to find the best-fit line to model the experimental results, 5 data points for zirconia and 4 data points for silicon nitride, against both Q_w' and a . The regression results were shown in Figs. 2 and 3, along with the best-fit lines.

3.1. Specific normal and tangential grinding forces, f_n and f_t , and grinding force ratio

Effect of Q_w . As the feed rate or Q_w increased, the grinding time is shortened and higher f_n and f_i are generated. At higher Q_w , the downward trend of

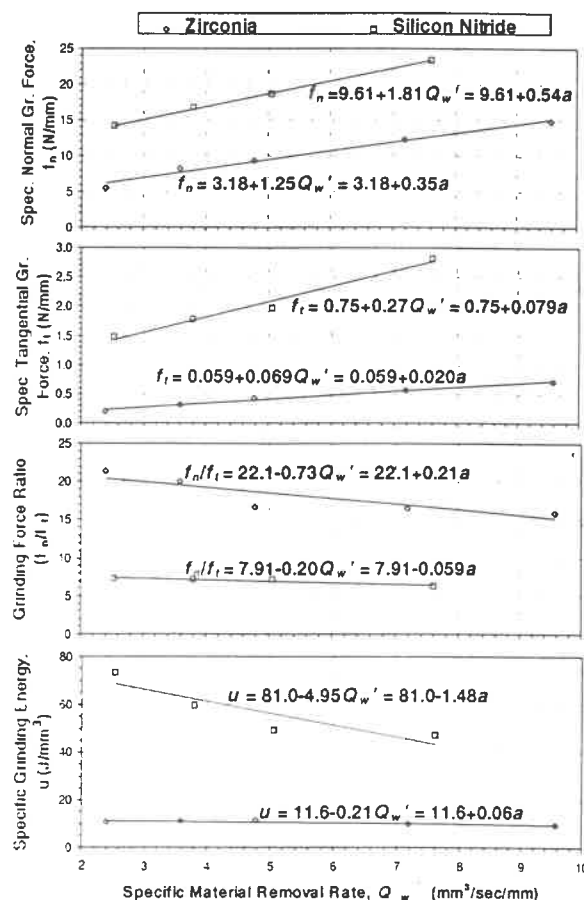


Fig. 2. The grinding forces, grinding force ratio, and specific grinding energy vs. specific material removal rate for cylindrical grinding of zirconia and silicon nitride.

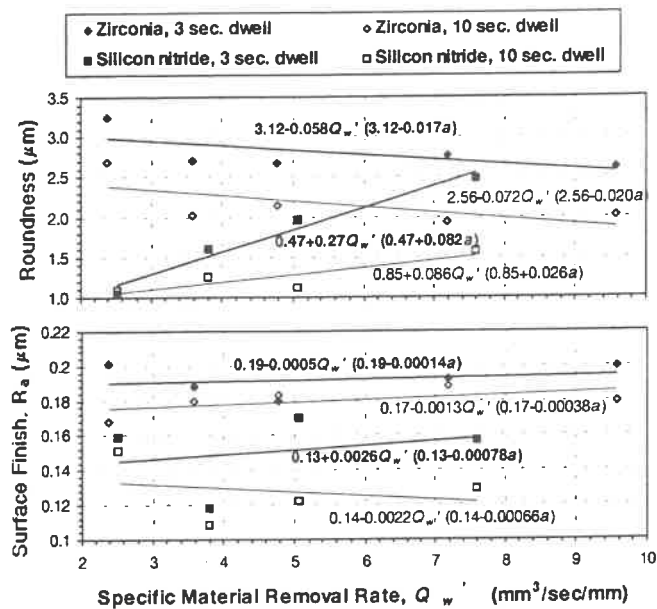


Fig. 3. The roundness and surface finish vs. specific material removal rate at 3 and 10 seconds dwell time for cylindrical grinding of zirconia and silicon nitride.

F_n/F_t illustrates that increasing friction occurs in both zirconia and silicon nitride grinding.

Effect of Work-Material The rate of increase of f_n and f_t for silicon nitride is higher than that of zirconia. The trend of increasing f_t for silicon nitride can explain why the test at 0.34 μm/s feed rate was failed due to the lack of motor power for the chuck.

3.2. Specific grinding energy, u

For silicon nitride, high Q_w' significantly reduces the u from 75 to 45 J/mm³. For zirconia, the u remains at about 10 J/mm³ across five material removal rates. The u for silicon nitride is considerably higher than that of zirconia.

3.3. Roundness

Figure 3 shows the roundness and surface finish vs. specific material removal rate (Q_w') at 3 and 10 seconds dwell time.

Effect of Q_w' The effects of Q_w' on roundness are different in silicon nitride and zirconia grinding. For silicon nitride, the roundness deteriorates at higher Q_w' . It may cause by the rapid increase in grinding forces. In contrast, the roundness of zirconia parts improves as the Q_w' increases from 2.4 to 3.5 mm³/s/mm.

Effect of Work-Material Silicon nitride parts, under the same truing and grinding conditions, have better surface finish than the zirconia parts.

Effect of Dwell Time Longer dwell time improves the roundness. For silicon nitride, the improvement of roundness is more significant at high Q_w' .

3.4. Surface Finish, R_a

Effect of Q_w' For zirconia, R_a remains in the 0.18 to 0.2 μm level at five different Q_w' . For silicon nitride, the overall trend shows the deterioration of surface finish at higher Q_w' .

Effect of Work-Material Under the same process conditions, better roundness can be found on silicon nitride parts.

Effect of Dwell Time Better surface finish can be seen on longer dwell time.

4. Concluding Remarks

In this paper, experimental results of high speed, high material removal rate grinding of silicon nitride and zirconia ceramics were presented. The benefits of high grinding wheel speed and high truing speed ratio were demonstrated to be beneficial to reduce the grinding forces and to increase the material removal rates. The increase in dwell time was also shown to be beneficial in terms of roundness and surface finish improvement.

The high grinding wheel surface speed not only generated lower grinding forces but also reduced the wear or improved the G -ratio when grinding ceramics. The follow-up experiments, results not included in this paper, showed that the G -ratio improves as the grinding wheel surface speed increases. Overall, this study showed that high speed grinding is beneficial to the grinding of ceramics.

Acknowledgment

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CHIP FORMATION AND SUB-SURFACE DAMAGE IN DUCTILE MODE GRINDING OF SEMICONDUCTORS

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Abstract

Single-crystalline silicon will also in the future maintain its outstanding position as substrate material for micro system technology. Due to the mechanical machining, damages are induced into the material that require a finishing of wafers by polishing and etching. Both processes are expensive and lead to a decreasing geometrical accuracy of the wafers. The quantities of surface quality and sub-surface damage for the processes preceding the polishing operation are therefore decisive factors for an economical wafer and part element production. One opportunity to improve the surface and sub-surface quality is presented by a plastic, ductile material removal. Maintaining the efficiency of the cutting grains and avoiding a filling-in of the grinding wheel is problematic when using finer grinding grains. References point to investigations of ductile grinding of ceramic materials [SEN95], which allow a ductile material removal when using coarse-grained grinding wheels. This paper examines whether these insights can also be applied to grinding of silicon.

Introduction

The mechanism of ductile material removal permits an almost damage-free surface generation. This again reduces the polishing ad-measure, so that polishing times are shortened and errors in shape occurring during the polishing process are minimized. This gains additional significance when considering increasing wafer diameters. The material removal of brittle-hard materials such as ceramics, glass, and silicon is subject of numerous scientific experiments and publications [BRI98, COO94, ENG97, HOL94, KOC91, MEN97]. For instance it is presumed to be certain that the removal of brittle-hard materials is based on the generation, spreading, and interlinkage of micro-cracks. Micro-cracks are produced when stress fields are induced into the material. Type, size, and shape of the emerging crack systems are related to the penetrating body geometry and the material properties with regard to the anisotropy and the position of the sliding system to the surface. The identification of working interrelationships between machining process, tool, material, and machine plays a major role in the realization of a ductile, i. e. crack-poor material removal.

To determine the boundary conditions for a ductile surface generation during abrasive stress, the Institute for Machine Tools and Factory Management (IWF), Technical University Berlin carries out scratch tests on silicon wafers. The objective of the investigation is to identify the parameters of engagement relevant for ductile surface generation. To be able to define guidelines for a ductile machining of silicon wafers, the parameters are varied in terms of tool geometry, cutting speed, feed, cooling lubrication, and material properties. Scratch forms are analyzed primarily to differentiate between plastic and brittle material behavior. The transition elastic-ductile to predominantly brittle removal behavior can be influenced by a number of parameters.

Experiments

To examine the removal mechanisms in ductile grinding and to analyze the relevant parameters of engagement influencing the removal mechanisms, scratch tests were conducted on a modified wafer grinding machine, type MPS-T500 by GMN, Nürnberg. Due to this procedure, the kinematics during scratching are equal to grinding kinematics. The emerging crack system depends in its type, size, and shape on the induced contact stress field. Therefore, the scratch tool has been changed with regard to the grain geometry in order to examine the influence of the cutting edge specification of the tools. Thus, pyramidal and roof-like single grain diamonds were used. In addition, the cutting speed was

varied from 15 – 35 m/s. In order to investigate the influence of the cooling lubrication, scratches were carried out dry and with water as cooling lubricant. Workpieces included both {100} and {111} silicon wafers. Tilting the scratch tool served to simulate a continuous feed.

The generated machining traces display four zones with different scratch characteristics (figure 1). The zones are primarily characterized by specific forms of micro-crack systems. Zone 1 shows a purely ductile, i. e. crack-free material removal. Subsequently, brittle removal mechanisms increase. Hence, zone 2 reveals first micro-cracks in the bottom of the scratch. Zone 3 has first micro-cracks occurring additionally on the scratch border. These cracks extend into the base material with increasing depth of scratch and eventually cause brittle, shell-like chippings (zone 4).

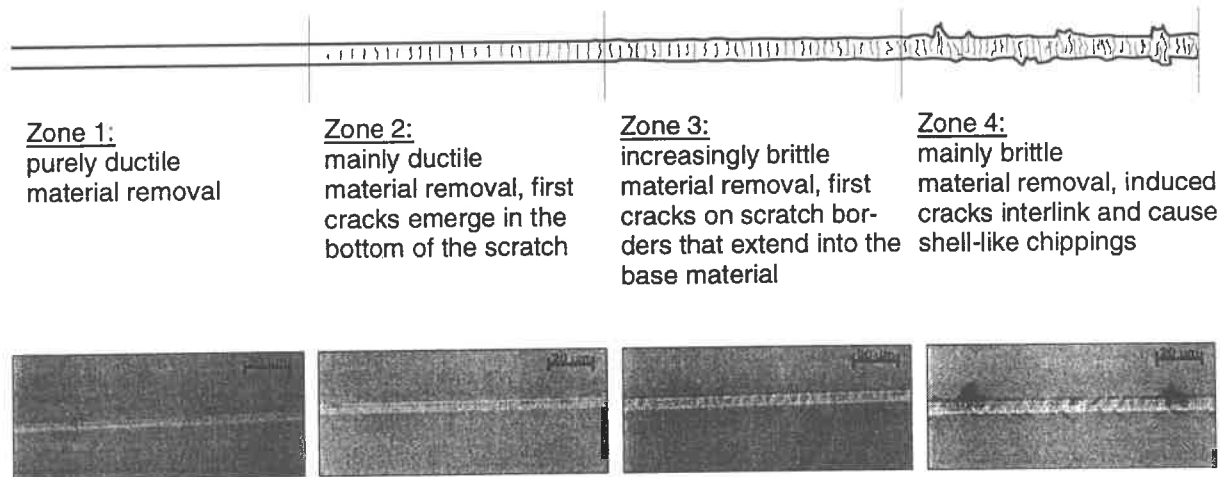


Figure 1: Scratch traces are divided into four zones that characterize the transition from purely ductile (crack-free) to predominantly brittle removal behavior.

Results

A light microscope serves to identify and characterize the single zones of scratch traces. The transitions between the single zones were geometrically measured. Hereby, a critical depth of scratch that defines the transition from one zone into the next could be assigned to the individual conditions of engagement. The first transitional zone is decisive for the ductile surface generation. It characterizes the change from purely ductile, crack-free material removal to mainly ductile removal with first recognizable cracks in the bottom of the scratch (figure 2).

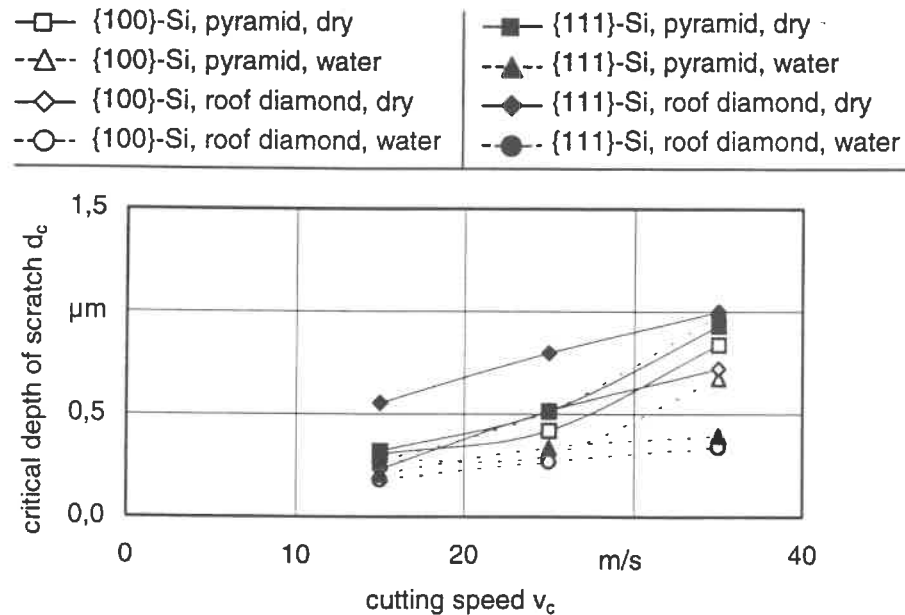


Figure 2: Development of critical depth of scratch of the first transitional zone in relation to the conditions of engagement.

The high uncertainty in measurement and the relatively small number of measurements do not allow safe statements in terms of absolute values. The most important influences can nevertheless be identified and qualitatively evaluated. Increasing the cutting speed in the tested zone from 15 – 35 m/s has a positive effect on the ductile surface generation, independent of the employed tool and workpiece specification and cooling lubrication conditions. The critical depths of scratch rise at an increase in cutting speed from 15 m/s to 35 m/s by values between 180 % and more than 360 %. The cutting speed is the parameter with the strongest influence on a ductile material removal. Using pyramidal diamonds leads to higher critical depths of scratch at otherwise equal parameters than during scratching with roof-like diamonds. Brittle removal mechanisms start earlier when using the roof-like, rather blunt tool. The cooling lubricant exerts an additional impact. Using water as cooling lubricant leads to an earlier transition to brittle shares in material removal. The influence of the crystal orientation of silicon wafers on the other hand is difficult to analyze. The critical depths of scratch of the {111} wafers tend to be higher than those of {100} silicon. This however does not include all conditions of engagement, so that the investigations conducted so far do not produce a cohesive result. The statements derived so far apply also mainly to the subsequent transitional zones (figure 3). Measured values disperse more strongly with increasing brittle fracture share, so that the statements become less certain.

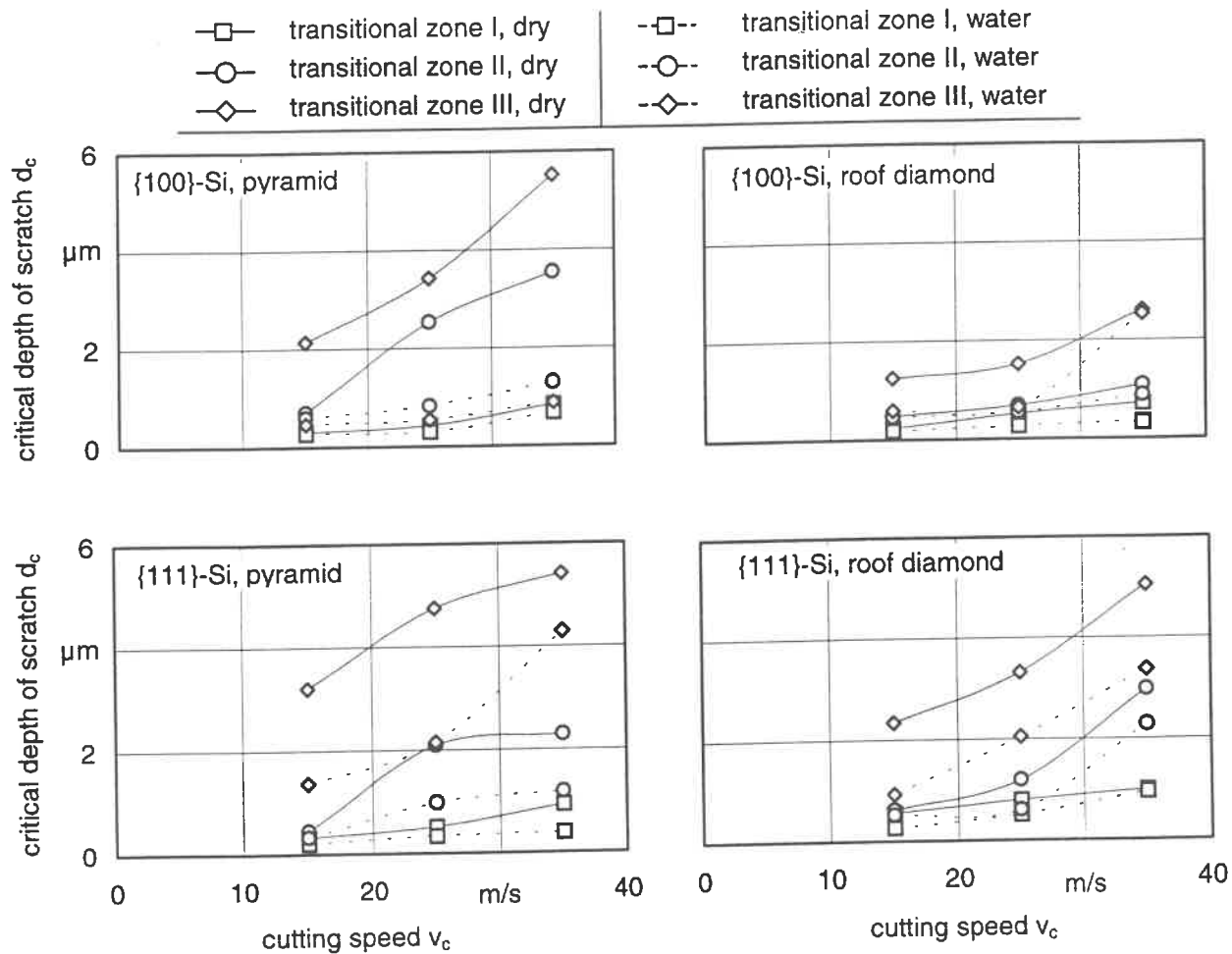


Figure 3: Development of critical depth of scratch in relation to the conditions of engagement.

The SEM pictures in figure 4 illustrate the difference in scratch trace development in relation to the tool in use. The roof-like tool causes a clearly delimited scratch. The wavy bottom of the scratch, which leads to the strongly dispersing measured values of depths of scratch, is plainly visible. The holes in the bottom of the scratch and next to the scratch trace indicate induced cracks that could not yet dissolve the material compound, but heavily disturb the crystal lattice. Using the roof-like tool already led to a contamination of the induced crack system and shell-like material chippings. The cracks extend far into the base material to the side of the scratch and damage the crystal lattice. The scratch trace is not so clearly developed in the preceding zones. The tool does not penetrate the material with one cutting edge, but with many small cutting edges.

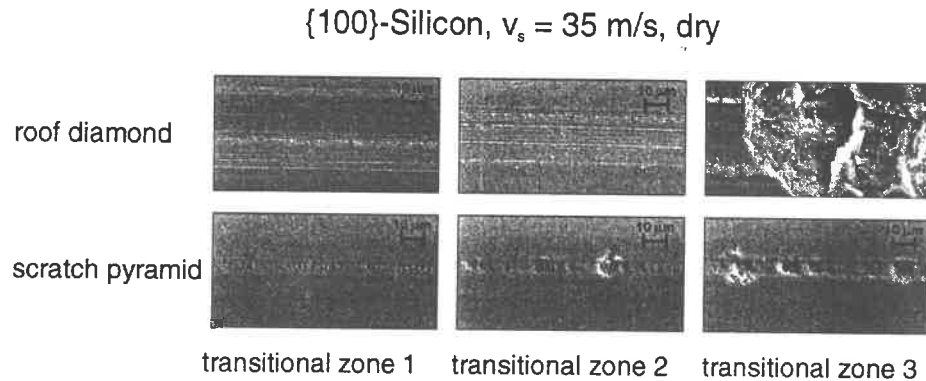


Figure 4: SEM-pictures illustrate the difference in scratch trace development in relation to the tool in use.

Conclusion

The following recommendations for grinding of silicon wafers can be derived from the results of the single grain scratch tests in order to realize a ductile material removal. If coarse grains are used for machining, sharp diamond grains are to be employed. The cutting speeds should be as high as possible. If there are no other technological reasons, cooling should be avoided. A temperature-resistant bond must therefore be selected in which the grains are firmly fixed. Grinding grains breaking out from the coating lead to an increase in depth of disturbance when they roll off on the workpiece surface. The bigger these grains are, the deeper the induced damage layer.

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New Abrasives Trends in Manufacturing of Silicon Wafers

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Approximately 150 million silicon wafers of all sizes were manufactured last year, representing a total turnover for the industry of \$6.2 billion in 1997 (Ref. 1). Wafer diameter has increased steadily from less than 50 mm in the 70's to 200 mm today (Figure 1). Originating from a silicon boule the same sequence of operations (Ref. 3 and 4) as described in Figure 2 has been used for wafer manufacturing up until the mid-90's. New productivity gains and technical challenges associated with the transition to 0.18 μm linewidth and/or 300 mm wafer diameter have now introduced new process choices for wafer manufacturers: wire sawing vs. ID sawing, grinding vs. lapping, double side vs. single side polishing (Ref. 5).

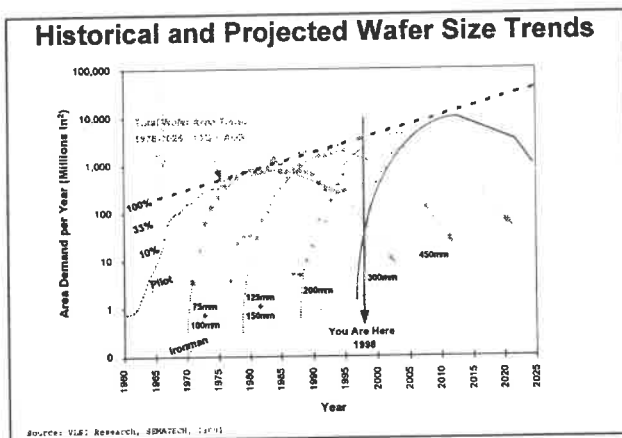


Figure 1: Wafer size evolution (from Ref. 2).

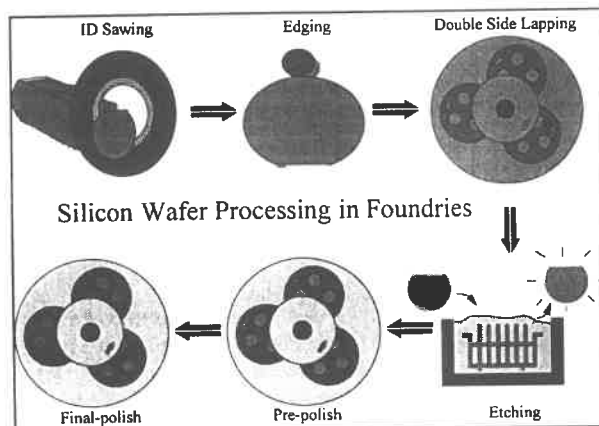


Figure 2: Conventional Si wafer process flow.

Sawing Processes: Most silicon wafers are grown using the Czochralski crystal pulling technique in which a seed crystal is dipped into molten silicon and slowly raised while rotating. This process creates a boule which is then cylindrically ground using a diamond abrasive cup wheel (Figure 3). Generating silicon wafers requires a slicing operation traditionally using an ID saw or more recently a wire saw. ID sawing is an interesting process (Ref. 6) in which the boule is introduced to the Inner Diameter (ID) of a thin sheet of high-strength steel held in tension on its outside diameter. The ID saw blade, which has been electroplated with diamond abrasive, rotates between 15 to 20 m/s while the boule is slowly raised to generate one wafer at a time.



Figure 3: Abrasive products used in Si wafers mfg.



Figure 4: Strasbaugh Model 7AF.

Wire Sawing Process: The concept behind wire sawing has been around for more than two decades (Refs. 7, 8 and 9) but it is only recently that it has been broadly applied to the manufacturing of silicon wafers. Wire sawing (Refs. 10, 11 and 12) consists of using one long steel wire wrapped several hundred times around grooved polyurethane guides. Loose silicon carbide (SiC) abrasive, typically 600-800 mesh suspended in a slurry is applied to the wire while the boule is plunged into the web. A thin ($\varnothing 160\text{-}250\text{ }\mu\text{m}$) and long wire ($>100\text{ km}$) moves between 5 and 15 m/s while slowly plunging down the silicon boule. Applicable for all wafer diameter but mostly used for 200, 300 or even 400 mm wafers (Ref. 13) the wire sawing process brings important productivity advantages by allowing the simultaneous slicing of up to a 1,000 wafers in 10 to 20 hours for the most advanced double table machines, compared with a single wafer process with ID sawing operations. The main drawback of wire sawing is its tendency to induce waviness with wavelength in the range of 0.5 to 30 mm, as seen with a Magic Mirror in Figure 5 (Ref. 14). On the other hand, waviness is not an issue for ID sawing especially when grinding is included during sawing. Table 1 summarizes both processes.

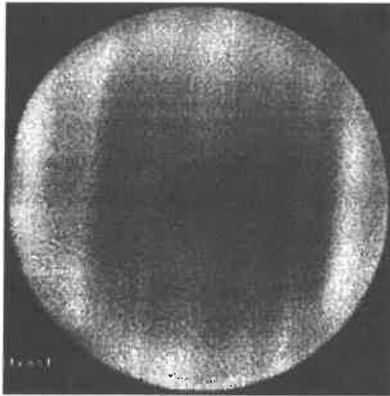


Fig. 5: Waviness after wire sawing.

Property	Wire saw	ID saw
Cutting Method	FAM / lapping	ploughing / grinding
Typical cut surface features	Wire marks	chipping & fracture
Depth of damage	Uniform 5 to 15 μm	variable 10 to 30 μm
Productivity	15-25 K.cm ² /hr	2-5 K.cm ² /hr
Wafer(s) per run	200 to 1,000 wafers	one wafer
Kerf loss	210-250 μm	250 to 400 μm
Min. thickness of wafers	200 μm	350 μm
Yield of 700 μm wafers	220 per 20 cm	180 per 20 cm

Table 1: Comparisons of various properties between wire saw and ID saw, from Ref. 14 and discussions with Andy Micham.

Wafer Thinning Process: Since the wafers coming out of the slicing operation do not typically have a satisfactory geometry, the slicing process is followed by one or several edge rounding operations (to chamfer the edge of the wafers using metal bonded diamond wheels) as well as a wafer thinning operation: double side lapping or more recently double side grinding. Wafer thinning improves flatness, bow, warp and total thickness variation (TTV) while reducing the amount of sub-surface damage (SSD) resulting from the slicing process. The most common form of wafer thinning is a double side lapping operation (Ref. 15) in which a batch of wafers (say 5 carriers with 4 wafers each for a total of 20 wafers) is manually loaded into a 1.5 or 2 meter diameter lapping machine. A loose aluminum oxide (around $\varnothing 10\text{ }\mu\text{m}$) slurry is then injected into the lapping machine which slowly rotates while removing silicon at rates of several microns per minute. This process tends to be fairly labor intensive.

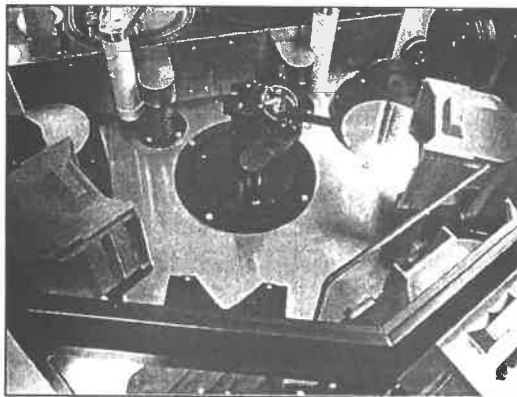


Figure 6: Cassette to cassette automation.

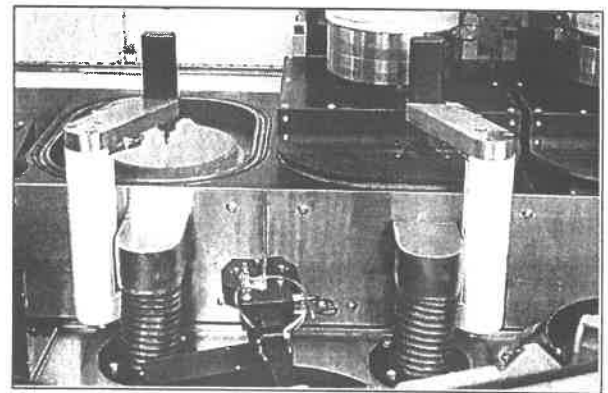


Figure 7: Measurement probe on the Strasbaugh 7AF.

A recent alternative to this double side lapping process consists of using a Fixed Abrasive Vertical Spindle (FAVS) approach, which easily lends itself to automation. These new developments rely on a technology which was

initially developed for the back-end of the semiconductor process, i.e. for wafer thinning operation (Ref. 16, 17 and 18) in wafer fabs just before dicing the individual dies. An example of a machine used for this FAVS process can be seen in Figure 4. It consists of a backgrinding machine with cassette-to-cassette automation as seen in Figure 6. Single wafer processing combined with single side grinding minimize the Total Thickness Variation (TTV) well into the sub-micron range. The key difference of this FAVS process compared with conventional back-grinding processes is in the need to use advanced technology wafer chucking techniques to eliminate the wire saw waviness. In process gauging (Figure 7) allows tight thickness control within the wafer and from wafer to wafer. An example of the performance data obtained with such a process vs. lapping is presented in Table 2. The current manufacturing status for 200 mm wafers (Ref. 19) is also presented in Table 3.

Parameter	Metrology	Double Side Lapping	Double side Grinding
Stock removed	ADE	70 μm	55 μm
Waviness	Magic Mirror	Not Visible	Not Visible
WTW Thick.	ADE	3-5 μm	1 μm
STIR	20 x 20 mm	$\sim 0.3 \mu\text{m}$	0.15 μm
Ra	Tencor P2	$> 4000 \text{ \AA}$	$< 100 \text{ \AA}$
Bow	ADE	6 μm	1.84 μm
Warp	ADE	N/A	8.44 μm

Table 2: Measurements on twenty 200mm ground wafers.

	200 mm
Tool	ADE 9300
GFLR	$\leq 1.5 \mu\text{m}$
GBIR/TTV	$\leq 2.5 \mu\text{m}$
SFQR (25 x 25)	$\leq 0.20 \mu\text{m}$
Bow	-14 μm to +10 μm
Warp	+10 μm to +40 μm

Table 3: Current manufacturing status for polished 200 mm wafers, Ref. 19.

Sub-Surface Damage: A technique to measure SSD of silicon wafers was developed by carefully polishing cross sections, etching them with a KOH solution and measuring the depth of damage on optical micrographs (Figure 8). Using this technique a study was conducted to measure SSD in silicon wafers at various stages of their manufacturing process (ref. 20). A total of 14 wafers were measured with three independent SSD measurement on each wafer with the results reported in Table 4.

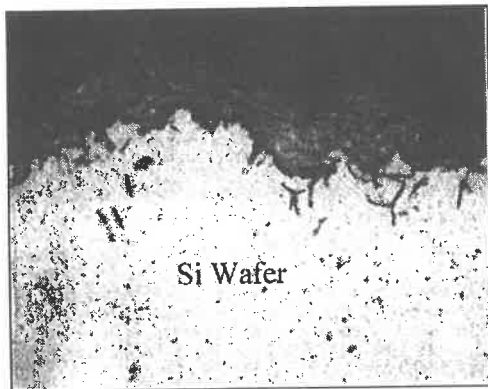


Figure 8: Edges of a wire sawn wafer after etching showing waviness (500X).

	# wafers	Avrg. SSD (μm)	Ra (μm)	Waviness (μm)
ID Sawing	2	10	3.5	15
Wire Sawing	3	18.3	6.0	N/A
Lapping	3	11.4	3.5	N/A
Etching	3	0	2.5	N/A
Polishing	3	0	0	N/A

Table 4: SSD measurements. Note that the damage after wire sawing is greater than after ID sawing.

Lundt et al have reported comparable SSD depth (Ref. 21): 18-25 μm for ID sawing with 50 μm average diamond size, 8-12 μm for lapping and as low as 1 μm for a grinding approach. Bismayuer et al (Ref. 22) also reported SSD measurements after ID sawing between 15 and 20 μm . Of course the exact process parameters influence the results as well as the specifics of the SSD measurement technique. For wafer thinning Kerstan et al (Ref. 23) have reported using grinding techniques to obtain surface roughness of 5-15 nm (measured with Atomic Force Microscopy - AFM) and SSD of 0.5-2 μm (measured with Scanning Infra-Red Depolarization - SIRD) with a 4,000 mesh wheel. Since grinding leads to better results than lapping it has been conceptually proposed (Ref. 24) to go even further and grind *both* sides of a silicon wafer at the same time to increase throughput even further.

Conclusions: As described in this paper wafer manufacturing processes have recently evolved from the conventional ID sawing/lapping/etching/polishing process flow to a new succession of operations which could only involve wire sawing, grinding and polishing. Such a simpler, cheaper and less labor intensive process would allow the wafer manufacturers to meet or exceed the requirements associated with 0.18 μm linewidth geometry or below as well as facilitate the transition to the 300 mm wafer generation.

Acknowledgments: The authors are indebted to Mr. Ron van de Merwe for developing the experimental technique to measure sub-surface damage of silicon wafers and measuring the various wafers mentioned in this paper. The authors would also like to thank Mr. Andy Micham for providing wafers at various stages of their manufacturing process and his multiple inputs in the writing of this paper.

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Design and characterization of a linear position transducer based on the Giant Magneto Resistance (GMR) effect.

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In this paper we describe the design and characterization of a linear position transducer based on the giant magneto resistance (GMR) effect. GMR is a relatively newly discovered phenomenon¹ observed in structures consisting of several magnetic and non-magnetic layers. Systems have been investigated during the last decade with reported $\Delta R/R$ ratios up to 65% at room temperatures². The GMR effect is roughly an order of magnitude stronger than the anisotropic magneto resistance (AMR) effect, which makes it very attractive for sensor designers^{3,4}.

Design of a linear position transducer

The GMR resistors used in our transducers were spin valve structures designed, manufactured and characterized by Nonvolatile Electronics Inc., Eden Prairie, MN. The principle of operation of these structures is depicted in Fig. 1.

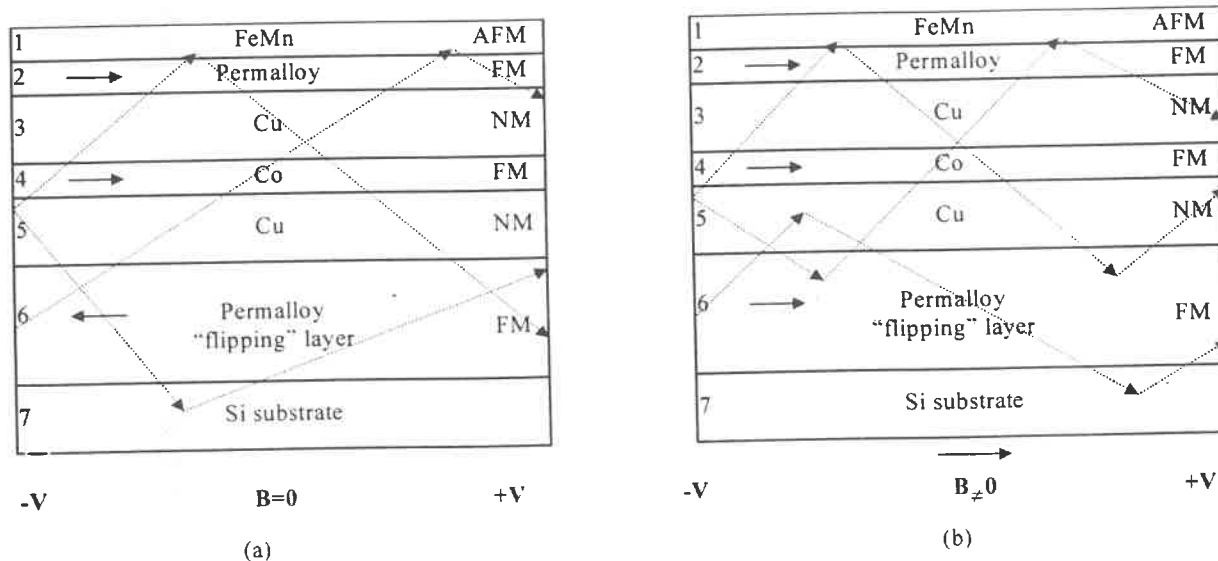


Fig. 1. Spin valve GMR structure in the absence of external magnetic field (a) and in the presence of external field (b). Arrows drawn in layers 2, 4 and 6 represent the magnetization of a particular layer. Broken arrows represent the trajectories of electrons when dc voltage is applied along layers. Case (a) is a "low resistance state": case (b) is a "high resistance state"

The occurrence of the GMR effect can be described as follows: A "soft" magnetic layer (6) made from permalloy is a "flipping" layer. Its magnetization is reversible under the application of an appropriate external magnetic field. This layer is strongly anti-ferromagnetically coupled with a "hard" magnetic layer (4) made from Co. This specific coupling is achieved by controlling the thickness of a "buffer" layer (5) made from Cu. (6) is also coupled, although somewhat weaker, with (2). This coupling is ferromagnetic. It is achieved again by controlling the thickness of another Cu buffer layer (3). The magnetically soft permalloy layer (2) is strongly coupled with the anti-ferromagnetic layer (1) made from FeMn. In the absence of an external magnetic field, anti-ferromagnetic coupling between the "flipping" layer (6) and the "pinning" layer (4) prevails and the magnetization of (6) is anti-parallel to the magnetization of (4) [Fig. 1 (a)]. Application of an external field [Fig. 1 (b)] flips the magnetization of (6). The

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field needed for this permutation is determined by the difference in energy between the anti-ferromagnetic and ferromagnetic coupling. This difference can be made quite small, so that the device will be sensitive to weak fields. This is in contrast to early GMR structures which required very strong magnetic fields.

When an electric field is applied, the spin-oriented conduction electrons travel the interlayer of a non-magnetic metal. Provided that thickness of this buffer layer is less than the coherence length, the conduction electrons arrive at the interface of the adjacent ferromagnetic layer still carrying their original spin orientation. When this adjacent magnetic layer is magnetized in a parallel manner, the arriving electrons have a high probability of entering the adjacent layer with negligible scattering. When the adjacent layer is magnetized in an anti-parallel manner, the arriving electrons suffer strong scattering at the interface. The GMR ratio is defined as:

$$K_{GMR} = \frac{\Delta R_{GMR}}{R} = \frac{R_{\uparrow} - R_{\downarrow}}{(R_{\uparrow} + R_{\downarrow})/2}, \quad (1)$$

where $R_{\uparrow}$ and $R_{\downarrow}$ are the resistance of the GMR element, when the layer with reversible magnetization is completely magnetized in each direction. $R = (R_{\uparrow} + R_{\downarrow})/2$ is the nominal resistance of the whole device.

The principle of displacement sensing is presented in Fig. 2. It shows the GMR stripe oriented along the axis of sensing. A movable magnet assembly is attached to the probe (not shown), the displacement of which is sensed.

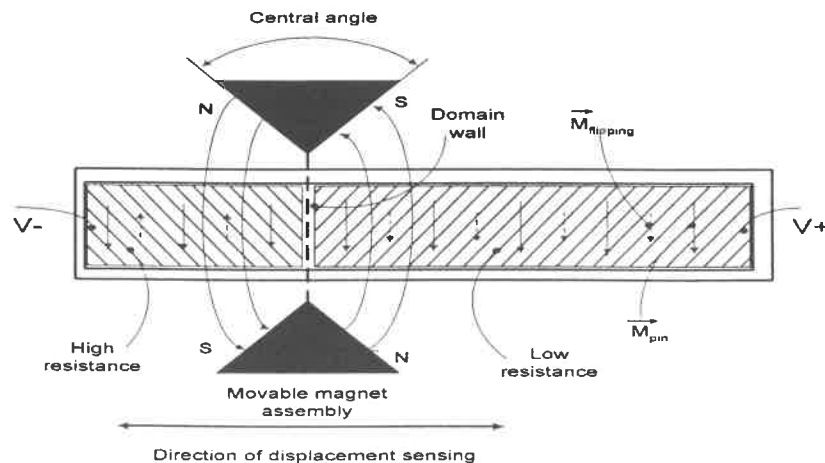


Fig. 2. Schematic principle of operation of the GMR-based linear position transducer. Shadowed area is the GMR stripe. Magnetization of pin layers and flipping layer is shown. Domain wall separates domains on the GMR stripe. If external voltage is applied, then left-hand domain has high resistance and right-hand domain has low resistance. Total resistance of the element is a linear function of magnet assembly position

Magnets create a magnetic field with a high gradient region located along the center line of the magnets. The configuration of the magnetic field is such that only one domain wall exists in the flipping layer of the GMR stripe. At the same time, the magnetic field is not strong enough to cause re-magnetization of the pinned layers. This magnetization was created during fabrication of the structure and remains intact for the life of the sensor. The domain wall separates two regions with anti-parallel magnetization. The left-hand domain has low resistance, because the magnetization in the flipping layer is parallel to the one in the pinned layers. The right-hand domain has high resistance, because the flipping layer and pinned layers have anti-parallel magnetization. When the magnet assembly has been displaced to the new position, the length of left-hand and right-hand domains changes correspondingly, which causes a change in the total resistance of the element. When the magnets are at the left-hand edge of the transducer, it has the highest resistance $R_{\uparrow}$. At the right-hand edge it has its lowest resistance $R_{\downarrow}$. In order to increase the signal-to-noise ratio and decrease susceptibility to temperature variations, four identical GMR resistors were configured into a Wheatstone bridge.

Model

Several physical effects determine the behavior of the transducer. Its transfer function can be written as $X_{out} = F(X_{in}, \text{temperature, direction, speed, etc.})$, where $F()$ itself is a parametric function of many design variables, such as GMR and AMR coefficients, nominal resistance, geometrical dimensions, magnetic field, etc.. A greatly simplified model which describes the most important features of the transducer correctly has been developed. It can be shown that due solely to the GMR effect the voltage measured across the bridge (V_{out}) is a linear function of displacement, measured from the center position (where bridge is balanced and $V_{out}=0$).

$$V_{out} = I_{in} \frac{R * K_{GMR}}{L} \Delta x, \quad (2)$$

where I_{in} is a constant current provided by a power supply and L is the length of the sensor. The situation is more complicated because conductors subjected to magnetic fields experience an anisotropic magneto resistive (AMR) effect. The following expression takes into account both GMR and AMR contributions to the sensor output:

$$V_{out} = I_{in} \frac{R * K_{GMR}}{L} \Delta x + (I_{in} \frac{R * K_{GMR}}{L} \Delta x) K_{AMR} \int_0^l B_y^2(l, \Delta x) dl, \quad (3)$$

where $K_{AMR} = \frac{\Delta R_{AMR}}{R}$ is usually of the order of 1-2%.

A complete model would also account for hysteresis, but this is hard to model quantitatively due to the complex nature of the effect. We expected hysteresis under assumption that GMR films although seeming continuous are actually a conglomerate of micro grains which switch their magnetization independently.

Results and Discussion

We used two precision linear position calibrators. One was designed and built by OPTRA, Inc. (Topsfield, MA). This unit used OPTRA's optical scale. Another one was designed and built by Federal Products Co (Providence, RI) and used a scale made by Micro-E, Inc. (Natic, MA). Both calibrators utilized precision air slides and provided accuracy of positioning on the 10 - 20 nanometer level. GMR resistors were fabricated by Nonvolatile Electronics, Inc. (Minneapolis, MN) on a 4" diameter Si wafer using sputtering techniques. The measured metrological parameters of the best transducers are summarized in the table below.

Resolution, 1 σ , (μ m)	Thermal coefficient. ppm/ $^{\circ}$ C	Deviation from linearity (μ m)			Hysteresis (μ m)		
@4 Hz update rate		23 mm range	8 mm range	1 mm range	23 mm range	8 mm range	1 mm range
0.2	2200	10	10	5	5	2	3

Fig. 3 shows typical response functions for the transducers utilizing various magnets. Greater angles correspond roughly to higher magnetic fields. Pronounced deviations from linearity at the ends of the range and hysteresis are in accord with the developed theoretical model. We found⁵ that devices equipped with 88 $^{\circ}$ central angle magnets had the best linearity and smallest hysteresis. In addition, some transducers (data not shown) demonstrated "steps" and "plateaus" in their response curves and had large hysteresis values associated with those abnormalities. We associated these anomalies with defects in GMR structures.

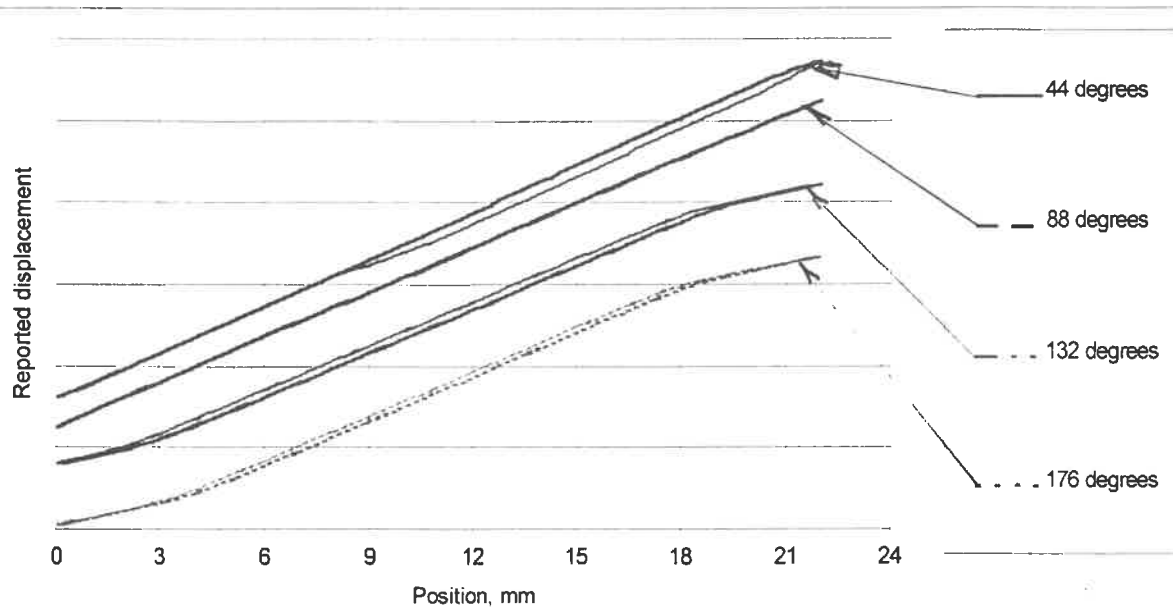


Fig.3. Experimental data from sensor #ZZZ with four magnets 44o, 88o, 132o and 176o central angle. Curves are off-set to provide better viewing. Curves order (from top down) corresponds to the angle sequence

Conclusions

It has been demonstrated that the GMR effect can be utilized for construction of linear displacement transducers. More than a dozen working prototypes were built and characterized.

Relaxation processes in the GMR structure and hysteresis have been observed. It is found that they can be minimized by choosing optimal magnetic fields. It is suggested that non-uniformity in thickness of the GMR layers as well as defects can be held responsible for observed non-linearity and noise. It is possible that ion-beam deposition would generate more uniform layers. Substantial thermal dependence of the output of the transducer has been observed. Some auto-compensation method should be developed to minimize thermal effects for industrial applications. . These issues should be addressed in the future work.

Acknowledgments

This work was done as a part of a consortium studying applications of GMR materials. Consortium members included Nonvolatile Electronics, Inc. (NVE), Honeywell, Inc., Naval Research Lab (NRL), and Hughes Research. These efforts were supported by the individual members of the Consortium and by DARPA. We appreciate the contributions of many people from Federal Products, and are also very grateful to Jim Daughton and Carl Smith from NVE, Hong Wan and Lucky Wiphanawsam from Honeywell, Gary Prinz, Peter Lubitz and Mike Miller from NRL and Stu Smith from UNCC for useful discussions on all stages of the work.

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Combining Pneumatic and Optical Vibration Isolation for Optimal Performance of an Artificial Star

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Keywords: star tracking telescope, pneumatic vibration isolation, optical feedback vibration isolation, automatic leveling

Gravity Probe B (GPB) is a satellite borne science experiment that will *measure* the distortion of space-time predicted by Einstein's General Theory of Relativity. It will do this by comparing the change in orientation of very precise gyroscopes as they undergo relativistic drift, to the reference direction of a guide star. Two predicted changes, called the geodetic and frame-dragging effects, are 6.6 arcseconds per year and 42 milliarcseconds per year respectively. The on-board telescope used to measure the reference direction to the guide star is accurate to 0.1 milliarcsecond, or 1/36,000,000 degree (Figure 1) [1].

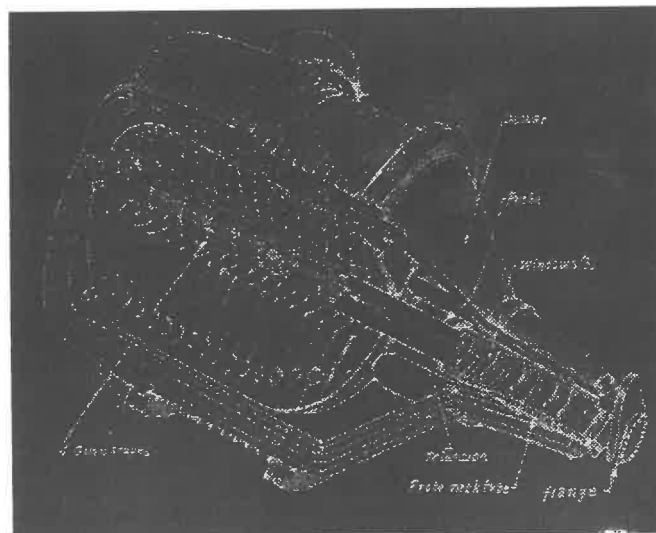


Figure 1 Inside the GPB satellite are gyroscopes and a star tracker telescope. The telescope provides a line-of-sight reference direction to a stationary guide star.

During telescope development, performance is tested using artificial star 2 (AS2) to simulate the guide star. AS2 is a large freestanding piece of equipment, including a mechanical vibration isolation system to attenuate floor vibration. The telescope may be mounted within AS2 and tested in a quiet mechanical environment. AS2 reaches extremely precise performance in testing telescope acquisition range, readout scale factor in the tracking range, and thermomechanical effects of cooling the telescope to 1.8 K. The very accurate AS2 is the primary means of testing the telescope during development.

After satellite integration, with the telescope mounted within the science instrument "probe" subsystem in the satellite, the complete star tracking system performance is tested again using artificial star 3 (AS3) [2]. AS3 must be a portable device, able to be mounted atop the satellite, and project its 6 inch diameter laser light beam into the telescope within. Sensitive beam formation and scan actuation optics are required to form a beam of sufficient

wavefront quality, and to steer the beam (Figure 2). AS3 can scan this light beam over a 2 arcminute range with 1 milliarcsecond resolution and 10-50 marcsec accuracy. Also, since it is impractical to remove seismic floor vibration from the satellite system in the assembly facility, AS3 must have a means to compensate for mechanical vibration of 0.3 arcsecond peak-to-peak beam angle between AS3 and the target telescope, at frequencies below 200 Hz. Therefore AS3 includes an active optical vibration isolation system to cancel this jitter and maintain the AS3 beam scan angle constant, relative to the telescope. The jitter cancellation system necessitates a structure insensitive to vibration with high $Q = \text{stiffness/mass}$. Optical feedback from a mirror on the front of the telescope is used to sense vibration between the light beam and the telescope. This signal is fed through a high speed controller into a piezoelectric mounted light weight mirror to optically cancel vibration in real time. The jitter mirror suspension has a natural frequency above 2 KHz. For a control bandwidth of 200 Hz, the plant (the mirror dynamics) can be considered a constant. Feedback is therefore pure integral control. The closed loop natural behavior has a real root, the magnitude of which is proportional to the gain.

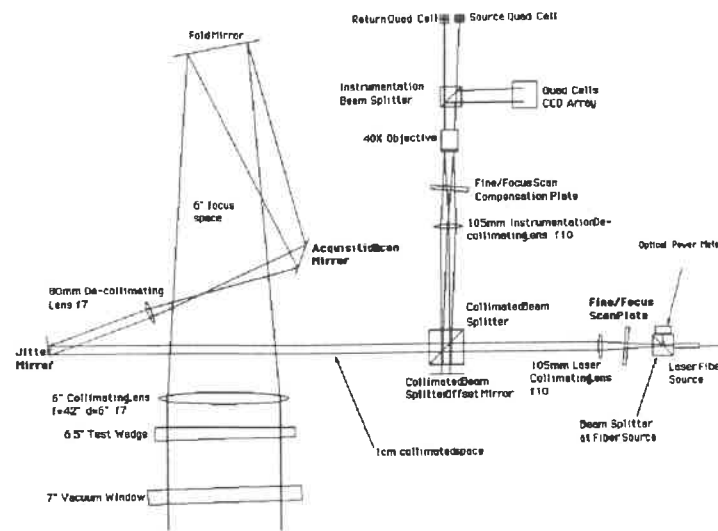


Figure 2 Diagram of the optics path of AS3, including beam formation, scan, and jitter instrumentation optics.

AS3 is used to verify that the GPB telescope meets the following NASA specifications:

- **Telescope Strehl Ratio:** The total Strehl Ratio of the telescope system (including the windows) and AS3 ($SR_{\text{telescope-windows+AS3}}$) is the ratio of the measured slope data, divided by the calculated ideal slope data. We can calculate SR_{AS3} using the equation

$$SR_{\text{AS3}} = \exp[-(2\pi\phi)^2],$$

where ϕ is the AS3 rms wavefront error measured using the Zygo interferometer (An AS3 rms wavefront error of 1/20 wave, about 1/4 wave peak-to-valley, gives an acceptable $SR_{\text{AS3}} = 0.9$). Knowing SR_{AS3} , we can determine $SR_{\text{telescope-windows}}$ using the equation

$$SR_{\text{telescope-windows}} = SR_{\text{telescope-windows+AS3}} / SR_{\text{AS3}}.$$

- **Telescope Scale Factor:** The telescope readout scale factor ($SF_{\text{telescope}}$) is defined as the slope of the output signal near the origin:

$$SF_{\text{telescope}} = [\Delta(I_- - I_+) / (I_- + I_+)] / \Delta\theta.$$

where $(I_- - I_+) / (I_- + I_+)$ is the beam power normalized telescope angle readout, and θ is the angle of the beam entering the telescope (Figure 3). SF is determined over the 1 arcsec linear tracking range. To measure $SF_{\text{telescope}}$ to within $\pm 20\%$, we must know θ to within $\pm 20\%$. This is the allowable error on the AS3 scan angle readout.

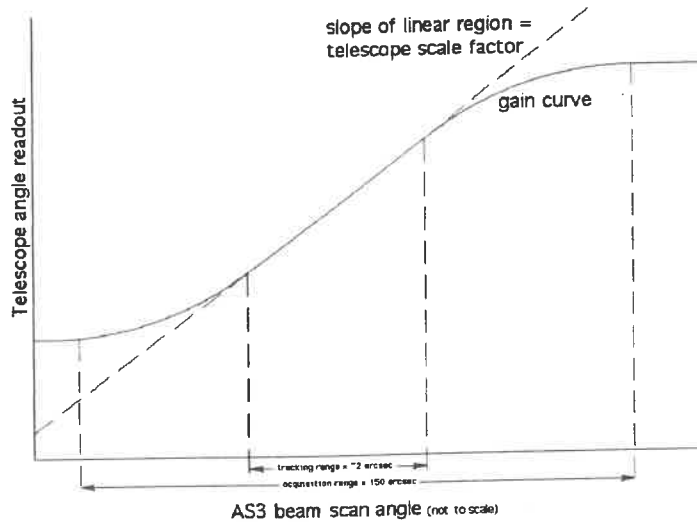


Figure 3 Graph of telescope scale factor.

- Telescope Optical Transmission: AS3 has an on-line means to measure beam power. Providing a beam power measurement to GPB will enable calculation of the total optical transmission through the probe.

To be able to test NASA telescope requirements, AS3 must meet a number of performance specifications itself, including beam wavefront quality, collimation, beam power stability, scan angle accuracy, optical jitter removal, as well as overall system readiness. A few tools are useful during AS3 development: a Zygo interferometer to measure beam wavefront, focus, and calibrate scan angle; a beam profiler to measure beam optic power; and a prototype GPB telescope to mimic the flight telescope while developing test procedures. A mechanically quiet environment is necessary during these tests to observe noise levels of AS3 subsystems with an absence of background noise. With a vibration isolated development environment, AS3 may be refined, beyond the ability to merely meet NASA specifications, to perform telescope tests with precision approaching that of AS2. By isolating the system during development we may be able to reduce disturbance to the threshold of optical, electrical, and mechanical performance of AS3. Finally, to be able to test AS3 optical jitter removal, we need to be able to approximate the mechanical vibration that we expect to see during the satellite test.

AS3 and the assorted testing tools (interferometer, beam profiler, prototype telescope, etc.) require a vibration environment during development that is quiet enough to give good SNR on performance threshold measurements. To meet satellite test requirements, we want an error on beam scan angle of about 1 or 5 in 100. Over a tracking scan range of 2000 marcsec, we desire a resolution of 20 to 100 marcsec. As a rule of thumb we are trying to approach a 1 marcsec beam scan angle resolution on all individual AS3 subsystems, with a cumulative total beam scan angle error closer to 10 marcsec. To reach scan resolution/error in the tens or marcsec range, we will require the interferometer to measure scan angle to 0.1 marcsec resolution. This is a measurement requiring a mechanically quiet environment. For the prototype telescope readout to be stable to a marcsec, we desire the jitter between AS3 and the telescope to be less than 1 marcsec. AS3 must be kept level in the presence of step disturbances and floor vibration. Tilting of the structure causes observable deformation of the optic path, turning the light beam relative to the telescope. Finally, vibration isolation needs to be de-tunable to match the vibration expected during the real telescope test on the satellite.

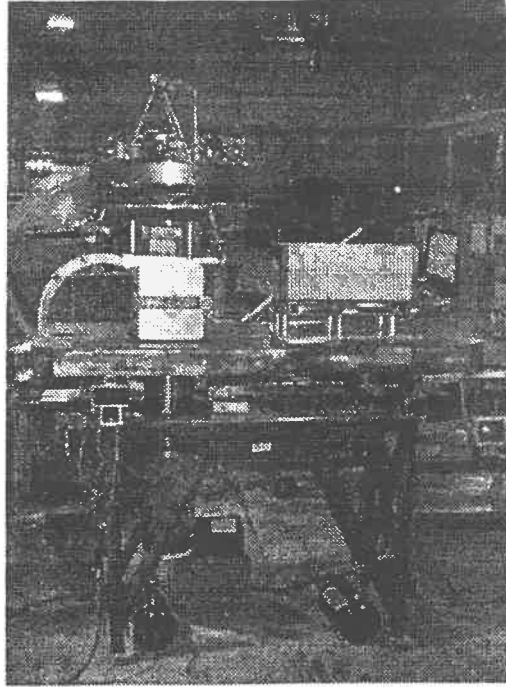


Figure 4 AS3 sits atop the prototype telescope on an optic table. The laser beam may be directed toward the telescope or a Zygo interferometer . The simulator assembly rests on three pneumatic isolators on a frame.

The vibration isolated assembly of a stiff optic table structural base, AS3, interferometer, beam profiler, and prototype telescope is called the "simulator" (Figure 4). AS3 is mounted above with its beam directed down into the prototype telescope. The beam may also be turned towards the interferometer or beam profiler. The 900 kg simulator assembly rests on three specially designed mounts (Figure 5). Each mount is comprised of an airspring, connected to a reservoir chamber. The airspring supports the simulator mass, and as the simulator moves, air travels back and forth between the chambers. A laminar flow restrictor is placed between the two chambers to act as a linear damper over a broad range of excitation amplitudes [3]. With geometry of the damper tuned and the volumetric ratio between the reservoir and airspring equal to 8:1, one can obtain critical damping of the sprung mass.

In addition to the passive pneumatic vibration isolation design, automatic height adjustment is achieved using a valve actuated by the height of the sprung mass. Gain is set by adjusting the valve arm length and the valve flow rate. Low valve actuation force minimizes structural path into the optic table. The addition of the air regulator provides active table leveling without having a significant effect on the vibration isolation response, as long as valve gain is set sufficiently low.

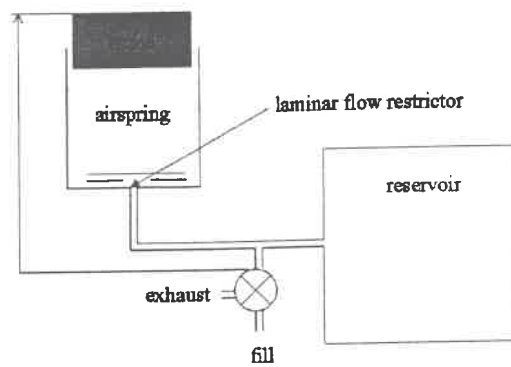


Figure 5 The simulator vibration isolation system is comprised of a passive laminar flow airspring system and an active table leveling system.

The pneumatic vibration isolation system minimizes disturbances entering the AS3/telescope system, but even these attenuated disturbances are observable with a telescope so precise. Complimentary methods are used to bring beam disturbance down to actuation resolution. Damped pneumatic vibration isolation is used with automatic height adjustment to minimize seismic vibration transmitted into the simulator, and an optical feedback closed loop control system actuates the AS3 light beam to maintain precisely the angle relative to the telescope (Figure 6).

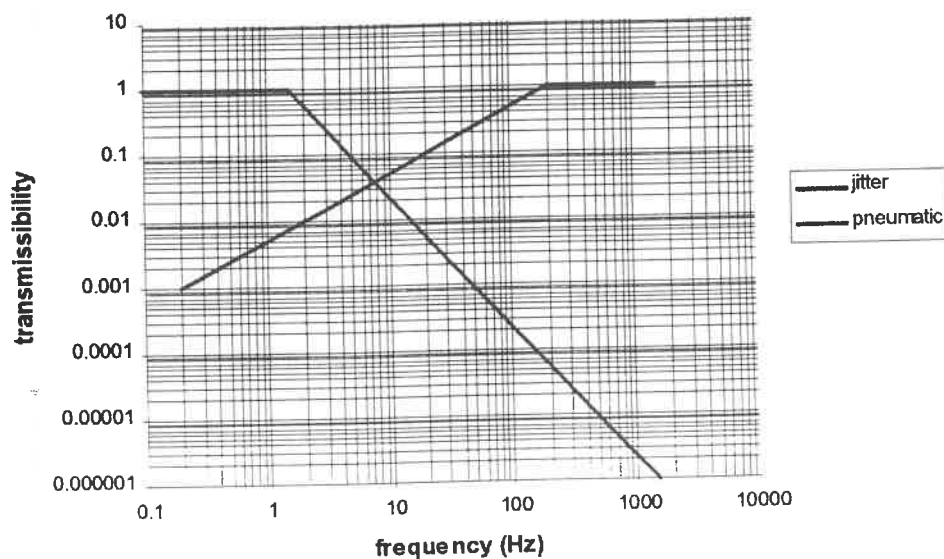


Figure 6 Transmissibility vs. frequency of the pneumatic isolation system and the optical jitter removal systems.

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